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Your Son?

Give Him a Man's Job!

New Roseland Pumping Station

By Edward K. Hammond

The pumping engines are placed in a pit above a basin into which the lake water flows by gravity. The four pumps are each of twenty-five million gallons capacity.

An increasing demand for water throughout the southern part of its territory has made it necessary for the City of Chicago to add to its facilities for supplying the requirements of that district. This need has been met by the construction of the new Roseland pumping station, now operating under part of its full rated capacity. The equipment is the most modern and, as a power-plant installation, presents many interesting features.

The building is of pressed brick and stone, with a tiled roof. Its main section is divided by a wall; the boiler room occupies the south end of the building, and the engine room is north of it. The building faces north and the engineers' offices are situated at both sides of the entrance.

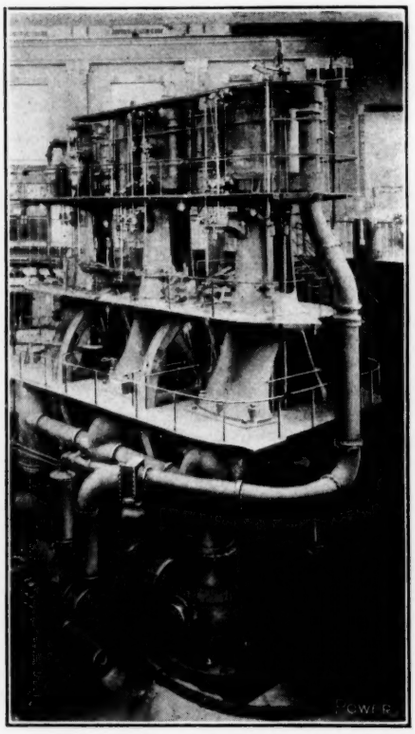


FIG. 1. WEST PUMPING ENGINE

PUMP ROOM

The pump-room floor is on the ground level, and the pumps are set in a circular pit in the center, Fig. 1, the walls and floor of which are concrete. A concrete basin is in the center of the engine pit, and at the top of a shaft at the end of the new southwest land and lake tunnel, which forms a connection through bedrock, between the station and the two cribs in Lake Michigan, from which the water is pumped. The depth of the engine pit is such that the level of the water in the basin is at Lake level, consequently the basin is filled by gravity.

The two pumping engines now in service have been installed in the southern half of the pit, leaving space for two similar units; arrangements have been made for installing a third engine, which will be of the same type as those already in use. These units consist of vertical 24, 48 and 70 by 60-in. triple-expansion Allis-Chalmers pumping engines, each having a capacity of 25,000,000 gal. of water per 24 hr., under a normal head of 140 ft., or 60.6 lb. per sq.in. The engines can pump against a maximum head of 200 ft., or 86.6 lb. per sq.in.; they are operated under 170 lb. steam pressure at the throttle.

The steam cylinders are steam jacketed and between the high- and intermediate-pressure and between the intermediate- and low-pressure cylinders are reheating receivers.

The basin from which the water is pumped furnishes the condensing water for the engines. Each exhaust pipe contains an oil separator and a primary heater between the engines and condensers. The surface condensers are designed to maintain a vacuum within 2 in. of barometric pressure. The condensed water goes to the hot well and is pumped back through the primary heater to the feed-water heater in the boiler room.

Each engine has a direct-driven air pump, feed pump and air compressor; there is also an independent air compressor and air pump, to provide the necessary compression and suction in the cylinders to start the engines. It has been found more convenient, however, to use water pressure for this purpose. Two pipes fitted with valves are connected to each water cylinder, which permit the water pressure to be turned on or released under any plunger, according to its position. The water pressure starts the pump when steam is turned on to take up the load.

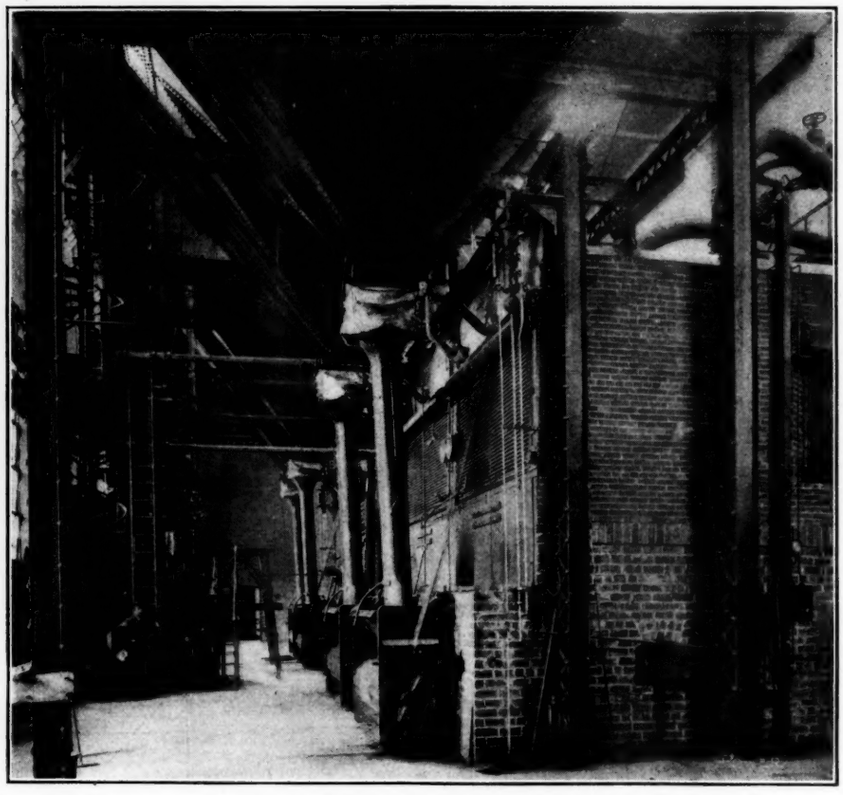


FIG. 2. VIEW OF THE BOILER ROOM

ELECTRIC GENERATORS

All electric current is furnished by two Curtis turbines directly connected to direct-current generators of 75 kw. capacity. In summer, the exhaust from these turbines goes to the feed-water heater in the engine room, but in winter it will be used in the building heating system.

Oiling System

The automatic oiling system with its gravity tanks makes it unnecessary to handle any oil in the plant after it has been deposited in the tanks in the oil room. These tanks are connected with two supply tanks, on the south wall of the engine room near the floor; the latter are kept filled by gravity feed. The oil passes through filters on its way to the supply tanks. Oil is pumped from the supply tanks to a third pair of tanks near the ceiling on the north wall of the boiler room. Connections lead from the lower of these two tanks to all bearings, guides, etc., and from the upper tank to all of the engine cylinders in the station. The supply tanks hold 5 bbl. and the gravity tanks 3 bbl. of oil.

The used oil runs through a system of drip pipes and is collected in reservoirs,

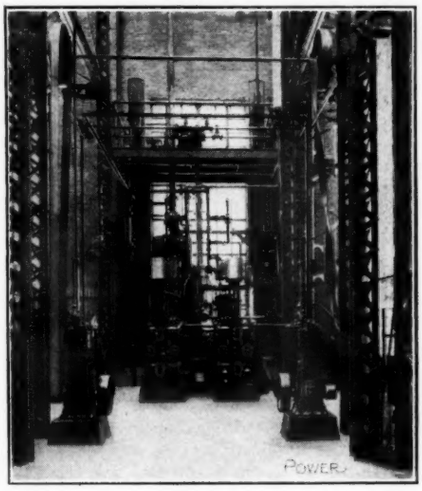


Fig. 3. Stoker Engines, Feed-water Heater and Feed Pump

from which it is pumped into oil filters on the wall of the engine room above the oil-supply tanks; the oil is pumped from the drip reservoirs into the filters and then drains to the supply tanks.

Boiler Room

The boiler room, Fig. 2, contains two batteries of two 300-hp. Edge Moor water-tube boilers, designed for 171 lb. steam pressure. The equipment is sufficient to take care of the four pumping units which will eventually be installed; only one battery is now used, but the third pump when installed, will require putting the second battery in service.

The boiler feed water is heated by the exhaust from all of the auxiliary engines in the plant.

A Link-Belt bucket conveyor runs over the boilers to deliver coal to the bunkers and down through the basement to take away the ashes. Coal is received at the plant on a spur from the C. & E. I. R.R. track, and is dumped into a hopper in the basement. After passing through a 40-ton crusher the coal is carried through the basement to the east end of the building, up through a sheet-metal casing surrounding the conveyor to the two double 170-ton bunkers over the boilers. One bunker feeds each boiler.

When the conveyor is to be used for removing ashes, the tripping mechanism is reversed. This throws out the trip over the coal bunkers and sets the one over the ash hopper at the west end of the

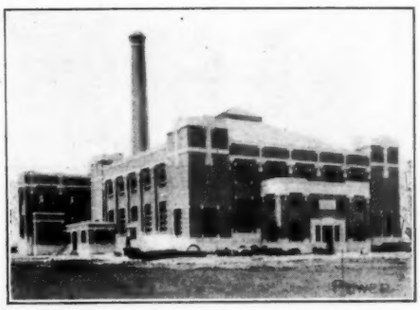


Fig. 4. New Roseland Pumping Station

building. The ashes are then taken up by the conveyor and carried around to this hopper, where they are dumped. The same siding over which coal is brought to the plant runs under this hopper, and the ashes are discharged into cars for removal, or they can be dropped through a second chute into wagons drawn up on a driveway beside the railroad siding. A system for wetting down the ashes before they are removed from under the grates has been installed.

The boilers are fitted with engine-driven chain-grate stokers, a single shaft extending over both batteries of boilers providing the drive for the eccentrics which actuate the rods connecting with the grates. Fig. 3 shows the arrangement of the engines which drive the stokers; also the feed-water heater and the feed-water pump.

The steam mains are piped from each boiler down through the floor near the north wall of the boiler room. Beneath the floor, these pipes make a bend through the wall into the south end of the engine room, where connection is made with a double header. All four boilers connect with each header, but the arrangement of valves is such that either header can be cut out when desired. The pumping engines take their steam from these headers through 6-in. pipes which are trapped before reaching the engines.

All piping is coupled by steel flanges welded to the pipe. The method of arrangement has been particularly well worked out, and an admirable combination of efficiency and simplicity have been secured.

Shafting and Belting Calculations

By Miles Sampson

The accompanying charts represent a very simple graphical solution of the problems which arise concerning shafting and belting. While presenting nothing new in theory, the application will be found very useful.

Shafting

The chart for shafting is based upon the commonly used formula for the strength of shafting:

$$Horsepower = \frac{(diameter\ of\ shaft)^3 \times r.p.m.}{constant}$$

In explanation of the constant, it may be noted that 100 is used for head-ends of shafts receiving large amounts of power. By largely increasing the size of

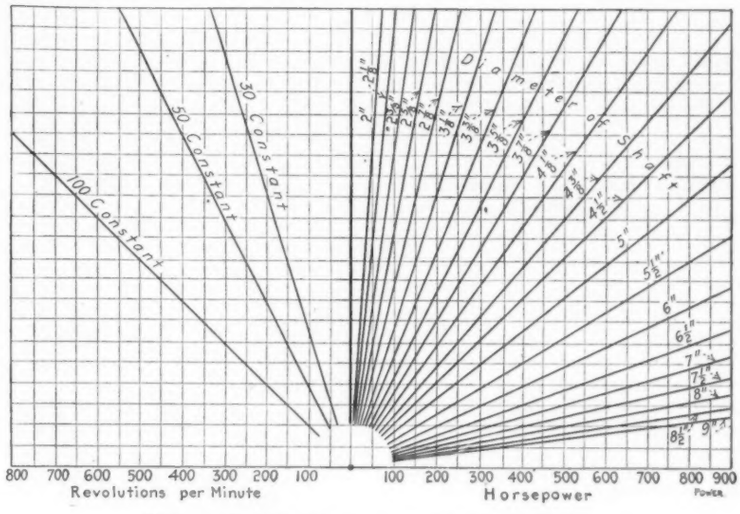


Fig. 1. Chart for Calculating Shafting

shaft necessary to carry a given power, it takes account of the large transverse stress due to the belt pull, and of the necessary stiffness. Any shaft carrying pulleys with belts over 6 to 8 in. wide should be figured using this constant. The constant 50 should be used to calculate countershafts driving ordinary machinery, and shafting carrying small belts. When

30 is used only the transmission of power should be expected of the shaft, no transverse stresses being considered.

The method of using the chart follows: On the scale representing the given speed follow vertically to the line denoting the proper constant, then across horizontally to the line representing the shafting size. The abscissa of this point on the horsepower scale gives the allowable load. Thus, at 300 r.p.m. a 3⅛-in. shaft carrying small belting (constant 50) would transmit 180 hp. Problems having different given conditions may be solved in a similar manner. For example—to carry 400 hp. at 400 r.p.m. a head-end shaft must be 4½ in. diameter.

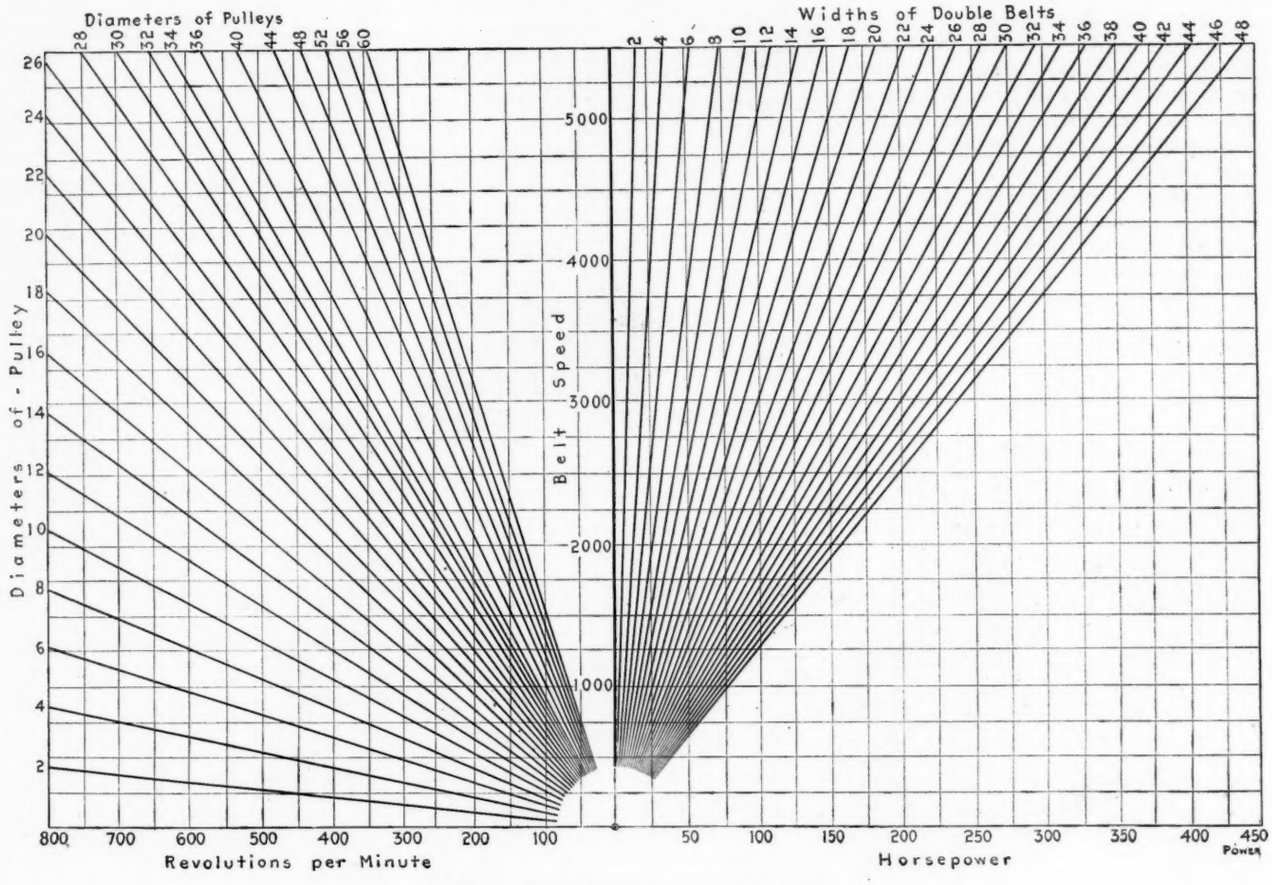


FIG. 2. CHART FOR DOUBLE BELTING

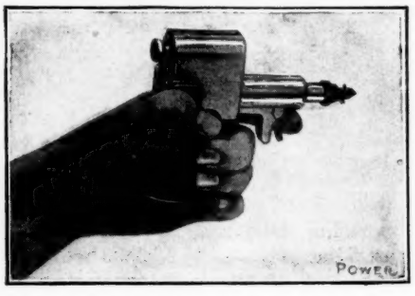
BELTING

The chart for belting is based upon a speed for double belts of 600 ft. per minute per horsepower per inch of width. This is conservative in the light of present practice, but represents a standard toward which belt users are tending. For very wide belts it may doubtless be decreased to 500 ft. per minute with perfect safety. For single belts the 600 becomes 1200, and for triple belts, 400.

For using the chart at any number of revolutions per minute follow vertically to the line representing the pulley diameter, which gives, on the vertical scale, the belt speed in feet per minute. Follow across horizontally to the belt width and down to the required horsepower. Thus at 400 r.p.m. on a 36-in. pulley (3750 ft. per minute) a 20-in. double belt will transmit 125 hp. Similarly, if a 30-in. belt is the maximum width available for transmitting 250 hp., a belt speed of 5000 ft. per minute is necessary; this requires a 60-in. pulley running 320 r.p.m.

Ever Ready Revolution Counter

The illustration shows a new type revolution counter particularly easy to handle, being shaped somewhat like a revolver and about the same size. The starting and stopping of the counting mechanism are not dependent on the degree of pressure with which the spindle is held against a revolving shaft but solely upon the operation of a trigger, thus eliminating chance for the counting to begin too soon or too late.



NEW REVOLUTION COUNTER

The spindle is held against the shaft and revolves freely with it, but the counter remains at zero until the trigger is pulled. The counting continues as long as the trigger is held, but stops the instant it is released. The figures, which are located in the head of the counter, can easily be seen and are quickly set back to zero. It counts forward whether the shaft is turning right or left. This counter is a recent product of the American Ever Ready Co., 304 Hudson St., New York City.

The engineering work in connection with the Panama-Pacific International Exposition at San Francisco, Calif., is proceeding rapidly. A department of mechanical and electrical engineering is at present engaged in studies for the underground distribution system and wiring systems of buildings for electric power. Careful detail estimates of the amount of power required for a great variety of purposes are necessary. Studies are being made for the selection of designs and sites for the service and other buildings.

Calculations of Pump Slippage

By W. G. Kirchoffer

Results of tests for slippage on several large pumping engines and a convenient diagram for determining the discharge of a pump with a given slippage.

It is a common practice among superintendents of water-works and engineers in charge of pumping plants to compute the discharge of their pumps on the basis of the actual displacement of the pump plungers. Such computations often lead to the statements made in the annual reports of the department that vast quantities of water are being used or wasted which cannot be accounted for. As a matter of fact, few, if any, pumps in this service ever discharge as much water as the displacement of the plungers would indicate.

The slippage of pumps depends upon many factors and cannot well be determined except by actual tests. In some cases where the water is pumped from or into reservoirs of known capacity, it is a comparatively easy matter to determine the actual capacity of the pump.

Besides the satisfaction of knowing that the consumption is not as large as supposed, such tests for slippage are of value in showing to the engineer the true condition of the pump. If such tests are undertaken, they should be made under practical operating conditions, for only then can the true slippage be determined. For example, suppose a pump is normally working against a pressure of 75 lb. per square inch, raising water out of a well or reservoir under a suction lift of 18 to 20 ft. The slippage under such conditions very likely would be much different from the slippage of a pump which might be tested by pumping out of a shallow tank or by allowing the water to discharge over a weir onto the ground.

In some tests which the writer recently made on pumping plants of various kinds, an opportunity was afforded for measuring the slippage of a number of those pumps. Table 1 gives the principal facts observed and recorded, but to make the results of greater use some explanation is necessary.

At Elkhorn and Platteville, Wis., the sources of supply are from deep wells which are pumped with an air lift. Here it was possible to fill the reservoirs previous to the test and then pump the water out, noting the strokes of the pump, the depth to which the water has been lowered and the time required to pump this amount. At Elkhorn, the direct-acting steam pump was short-stroking so that the stroke was only 8½ in. in place of 10 in. The displacement and slip given in the table are computed for an 8½-in. stroke, but the same figures for a 10-in. stroke would be 6.65 gal. and 22.4 per cent. slip. At Platteville, the displacement was based on the full stroke.

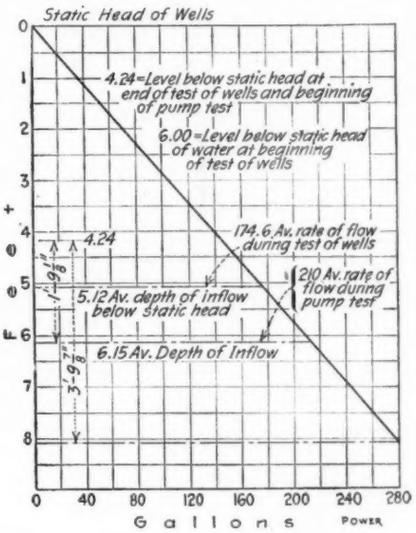

Static Head of Wells
4.24=Level below static head at end of test of wells and beginning of pump test
6.00=Level below static head of water at beginning of test of wells
174.6 Av. rate of flow during test of wells
210 Av. rate of flow during pump test
4.24
1' 9⅛"
5.12 Av. depth of inflow below static head
6.15 Av. Depth of Inflow
3' 9⅞"
0 40 80 120 160 200 240 280
Gallons
POWER

FIG. 1. METHOD OF CALCULATING FLOW OF WELLS

At Kaukauna, Grand Rapids and Watertown the supplies are obtained from flowing wells discharging into masonry reservoirs. At the two latter places the conditions were such that the flow of the wells was not affected by the level of the water in the reservoirs so that it was possible to gage the flow previous to the test, when the pumps were not operating, by noting the rise of the water in the reservoir in a given time. The actual quantity of water pumped during the test is the amount pumped from the reservoir as measured by the depth pumped plus the flow of the wells for the given time.

The supply at Kaukauna is from two artesian wells which discharge against the head of water in the reservoir. When the water was pumped down, the wells increased in flow, so it was necessary to determine not only the flow of the wells at any one level but the rate of increase of flow as the water in the reservoir receded due to pumping. The approximate static head of the wells was obtained from the engineer and a record of the time and depth of inflow was recorded for a period of 3 hr. 46 min. During this time the water level raised 1 ft. 9⅛ in., corresponding to a flow of 39,400 gal. in 226 min., or an average of 174.6 gal. per minute. At the end of the test for inflow, the pumps were started and operated for 1 hr. 45 min., during which time the water was lowered 3 ft. 9⅞ in.

By constructing a diagram as shown in Fig. 1 and assuming that this rate of increase would be uniform, which is fairly correct for artesian wells, and placing the average flow of 174.6 gal. at an average depth of the inflow, or at 5.12 ft. below the static head, it is possible to read from the diagram the average rate of inflow during the period of pumping, which is approximately 210 gal. per minute or a total inflow during the test of the pumps of 22,050 gal.

This test could have been made by the method used at Wausau, Wis., which was as follows: At Wausau one of the supplies is from a round well 36 ft. in diameter by 35 ft. deep, the water en-

TABLE 1. RECORD OF TESTS FOR PUMP SLIPPAGE

Location	Type of Pump	Size of Pump, Inches	Capacity of Pump, Gal. per Stroke	Diameter of Reservoir	Capacity per Foot of Depth	Depth Pumped out of Reservoir	Gallons of Water Pumped out of Reservoir	Flow of Wells during Test	Total Gallons Pumped	Duration of Test, Hr.-Min.	Str'kes of Pump	Actual Discharge per Stroke, Gal.	Slip in Per Cent. of Displacement
Elkhorn, Wis.	Compound duplex	8, 12&7x10	5.65	18′	1,904	16′– 3⅞″	31,038.0	0	31,038	2–13	6013	5.16	8.6
Kaukauna, Wis.	Compound duplex	14, 20&12x18	35.3	61.7′ 1–30′	22,400 5,199.57	1′– 9¾″	85,600	22,050	1,076,501	1–45	3212	31.92	9.5
Platteville, Wis.	Compound duplex	12, 18&10x12	16.32	1–29′	5,285.00	2′–10⅝″	29,635	0	29,635	2– 0	2336	12.68	22.3
Grand Rapdis, Wis.	Double-acting triplex	8x8	10.1	40′	9,424.5	4′– 0″	37,698	17,200	54,898	2–11½	6580	8.35	17.3
Wausau, Wis.	Crank and flywheel Compound duplex	14¼x24	64.12	36′	9,613.7	2′–10″	21,580	28,764	50,344	0–35	1016	49.52	22.8
Watertown, Wis.	Compound duplex	14, 20&12½x18	38.3	60′	21,054.0	1′– 4″	28,000	14,220	42,220	1– 0	1204	34.9	8.2

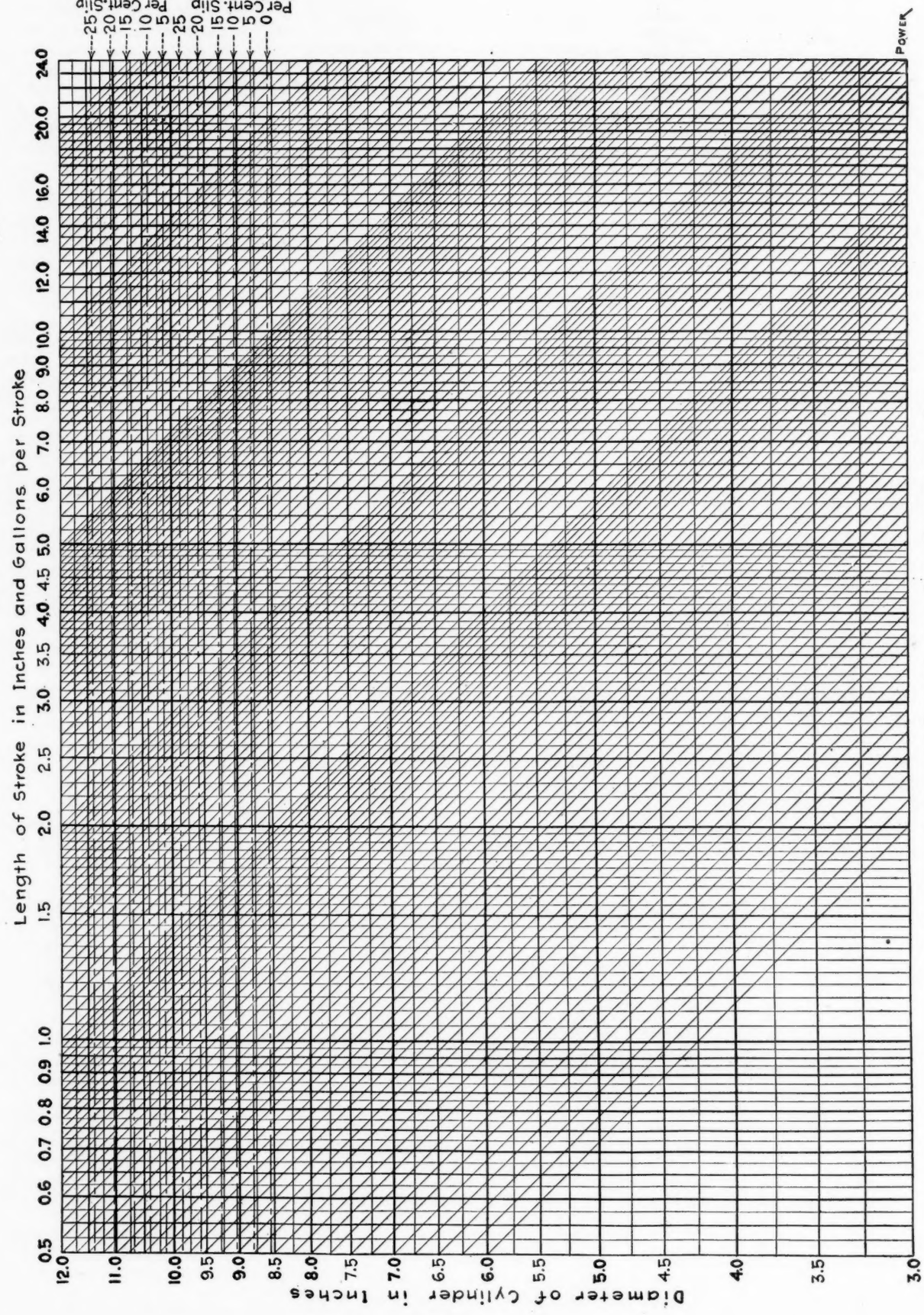

Per Cent. Slip
Duplex Pump
Double Acting
Per Cent. Slip
Triplex Pump
Single Acting
0 5 10 15 20 25
0 5 10 15 20 25
POWER
Length of Stroke in Inches and Gallons per Stroke
0.5 0.6 0.7 0.8 0.9 1.0 1.5 2.0 2.5 3.0 3.5 4.0 4.5 5.0 6.0 7.0 8.0 9.0 10.0 12.0 14.0 16.0 20.0 24.0
Diameter of Cylinder in Inches
12.0 11.0 10.0 9.5 9.0 8.5 8.0 7.5 7.0 6.5 6.0 5.5 5.0 4.5 4.0 3.5 3.0

FIG. 2. CHART FOR DETERMINING PUMPAGE FOR A GIVEN PERCENTAGE OF SLIP

tering through the open bottom. The water is pumped by two Holly-Gaskell direct-pressure crank and flywheel pumping engines. As there were no storage reservoirs and the pressure had to be kept up by constant pumping, it was necessary to make the test by observations upon this round well. As the flow of water came in through the bottom and increased more rapidly than was proportional to the lowering of the water, it was necessary to resort to other methods than those used at the other places mentioned. The rate of inflow for short periods of time, such as intervals of 5 min., no doubt increased fairly uniformly.

Proceeding on this assumption a hook gage was arranged in the well so as to have a long travel and readings on the level of the water in the well and the revolutions of the pump were taken every 5 min. for a period of 35 min. The period was short because the inflow soon become nearly equal to the pumpage and the high velocity made the water turbid. The pumping was continued for a few minutes after the pumping test was completed, or until the suction could be changed over to a driven well system. Then a record was made of the time it took the water to rise to each of the readings made when pumping out the well. A copy of this record for one of the pumps is given in Table 2. The quantity of water pumped during any 5-min. period is represented by

$$Q = 7613.7\ L\left(1 + \frac{t}{t'}\right)$$

in which t' must correspond to the time required for filling the well a corresponding distance of L ft. The last part of the formula within the brackets represents the flow of the well in the time t and for sake of clearness will be explained.

If a quantity of water flows into a well, or reservoir, represented by a depth L, which was observed as shown in the table, then the average flow per minute is 7613.7 L divided by t' minutes and the flow in the time t during which the pumping took place is t times this amount or

$$7613.7\ L\left(\frac{t}{t'}\right)$$

The 1 in the brackets represents the water in the well as storage which was pumped out during the time t.

The data given in Table 1 are also shown graphically in Fig. 2. The slip of the pumps was computed by the following formula:

$$S = \frac{Q - q}{Q} = \frac{64.1179\,R - 7613.7\,L\left(1 + \frac{t}{t'}\right)}{64.1179\,R}$$

where,

Q = Displacement of pump in gallons in time t;
R = Revolutions of pump in time t;
q = Actual discharge of pump in gallons;
L = Lowering of water in time t;
t' = Interval of time for well to fill up distance L which was pumped out in time t;
t = Interval of time in taking readings during pumping; it must be a constant interval;
S = Slip of pump in percentage of displacement;
64.1179 = Displacement in gallons for one revolution of pump;
7613.7 = Capacity in gallons for one foot of depth of reservoir;

$$Q = 64.1179\ R;$$

$$q = 7613.7\ L\left(1 + \frac{t}{t'}\right).$$

This test was applied to both pumps, one at a time. The average of a number of determinations of each was 22.8 per cent. for one and 22.66 per cent. for the other. In using a method of this kind extremely accurate readings must be taken; otherwise the results will be of little value.

Discussing the results of all the tests briefly, it is the belief of the writer that the results found at Elkhorn, Kaukauna and Watertown show a fair average working condition of the pumps, considering that they have been in service from five to ten years. At Platteville the high percentage of slip, no doubt, was due in part to short-stroking the pump. At Grand Rapids, the pumps were not kept in a good condition from a mechanical standpoint. At Wausau, the high percentage of slip was due in part to the age of the pumps (having been in use about 25 years) and to a heavy suction lift.

In making tests of this kind, or in any engineering work connected with pumps, it is often convenient to have at hand some ready means of determining the displacement of pumps and the actual capacity at differtnt percentages of slip. For this purpose the writer has devised the diagram shown in Fig. 2. The use of this diagram perhaps can be explained best by a few examples.

EXAMPLE—What is the capacity of a 6x8-in. double-acting duplex pump, assuming no slip?

From the left-hand side of the diagram follow the horizontal line opposite 6 in. until it intersects the vertical line through 8 in., the length of the stroke; then follow the diagonal line from this intersection upward to the left until it intersects the horizontal dot-and-dash line of zero slip; now follow the vertical line to the top of the diagram and read off 3.92 gal. per stroke. If the capacity is desired when the slip is 5 per cent., follow the diagonal line upward to the horizontal dot-and-dash line representing 5 per cent. and read off as before at the top of the diagram 3.7 gal. per stroke. Similar determinations can be made by this diagram for any percentage of slip up to 25.

If the pump were a 6x8-in. single-acting triplex the same procedure would be followed out except that in following upward on the diagonal line, proceed farther up until the line of zero slip is intersected for these pumps. If the size and type of pump are known and the actual discharge has been determined by test, such as described in the first part of this article, the percentage of slip may be readily read from the diagram by following down a vertical line from the actual capacity until it intersects the diagonal line from the intersection of the horizontal and vertical line representing the diameter and the stroke of the pump.

The diagram can also be used for determining the discharge of the pump per minute when the number of strokes is known. For example, take the same size of pump used in the previous problem and assume that it is making 40 strokes per minute. Forty times eight equals 320. The scales being logarithmic, this may be read as 3.2 on the diagram.

TABLE 2. RECORD OF TEST FOR SLIPPAGE OF PUMP AT WAUSAU, WIS.

Time of Pumipng, Read Down	Time Interval of Pumping, t	Time Filling Reservoir, Read Up	Time Interval of Filling, t'	Gage Reading in Well	Lowering in Feet, L	Reading of Revolution Recorder	Revolutions per Interval of 5 Minutes	$L\left(1 + \frac{t}{t'}\right)$	Displacement of Pump, Gallons	Actual Discharge, Gallons	Per Cent. Slip
8:40	5	10: 9:10	.....	7.970	.....	3249	...	.....	.....	.....	....
45	5	9:58:38	10.32	7.375	0.595	3388	139	0.877	8,915.0	6680.00	25
30	5	9:51:15	7–23	6.780	0.595	3541	153	0.998	9,815.0	7598.48	22.5
55	5	9:46:07	5–08	6.195	0.585	3723	182	1.155	11,665.0	8793.84	24.6
9:00	5	9:43:01	3–06	5.745	0.45	3905	182	1.174	11,665.0	8938.50	23.4
05	5	9:40:57	2–04	5.395	0.35	4086	181	1.197	11,600.9	9113.61	21.5
10	5	9:39:35	1–22	5.135	0.26	4265	179	1.211	11,476.0	9220.20	19.7

Average, 22.8

From the intersection of the horizontal line through a diameter of 6 in. and the vertical line through 3.2 in., proceed as before and read the true displacement at 156 (actual reading 1.56) gallons per minute at the top of the diagram. Where the intersection of the horizontal and vertical lines come above the horizontal lines of slip, such as would be the case for a 10x10-in. cylinder, the diagonal line is followed downward to the right in place of upward to the left. For the larger sizes of pumps represented on the diagram where the intersections of the diagonals with the slip line would come off the diagram, such as 10x20 in., the capacity may be determined by using a 2-in. stroke in place of 20 and multiplying the result by 10.

The diagram is also useful in finding what cylinders of different dimensions and strokes have the same capacity. In this connection it is interesting to note that cylinders of one-half the diameter and four times the stroke of a given cylinder have the same capacity; that is to say, a 12x1½-in. cylinder is equivalent to a 6x6-in. or a 3x24-in. cylinder in capacity per stroke.

This diagram is limited in diameters of cylinders between 3 and 12 in. and for strokes from 0.5 to 24 in., but by making use of its logarithmic properties, it may be used for any stroke. It is also limited to the use of double-acting duplex pumps and single-acting triplex pumps. Similar diagrams could be readily prepared for single-acting single-cylinder or for double-acting triplex pumps and for greater ranges of diameters.

Care and Management of Water Tube Boilers

By C. K. Heasley

Care should be exercised in providing boilers of sufficient capacity for the work required. Many steam plants equipped with water-tube boilers are so arranged that the engineer does not have the opportunity to properly care for the boilers. A sufficient boiler capacity should always be provided so that one boiler may be down for overhauling and washing out. The water-tube boiler should be washed out just as often as any other type and under ordinary conditions it should be done at least every 30 days, although local conditions will determine just how often in the individual case.

The following method of washing out horizontal water-tube boilers gives best results: First, the boiler should be cut out of battery and allowed to stand with the dampers closed until the pressure has been reduced to zero. Then, open the blowoff and allow the water to run out until the drums are empty. Remove the manhead and thoroughly wash the drum. Next allow the water to run out to the bottom level of the top row of tubes. Close the blowoff and remove the caps or handholes on the top row of tubes and pass the scraper or turbine or cleaner through each tube. Allow the water to run out so as to empty the second row of tubes. Repeat this until all of the tubes have thus been scraped. When the last or bottom row is cleaned, the boiler is empty and clean and the heat from the furnace walls, arches and baffles has not baked any of the mud onto the tubes.

Care should be taken to clean out the combustion chamber and the space underneath the steam drums, particularly between the tubes, where the ash and soot usually clinker together and fill up between the tubes at the rear end. It is just as important to keep the tubes clean externally as internally, for if the space between becomes filled up this localizes the heat and often is the direct cause of a ruptured tube. Care should be taken never to blow out between the tubes with wet steam. Always allow the steam to blow through the blow pipe for a sufficient length of time to allow all of the wet steam to escape. If steam with excess moisture is used for this purpose, it will dampen the soot and ash and clog up between the tubes. After the boiler is fired up, these will burn hard and become like clinkers, and it will be found very difficult to remove them, especially in boilers where the tubes are staggered.

In erecting boilers they should be set, if possible, at least 4 ft. apart, to allow of cleaning out between the tubes from the side by having side doors located between baffles. Many manufacturers do not provide openings necessary for the proper access to the boiler so as to care for and keep it in good condition. Engineers in charge of the erection of boiler plants should see that these provisions are made.

The mud drums and bottom nipples should be thoroughly scraped each time the boiler is washed out and coated with some good nonrusting paint. A good paint for this purpose and one that can be relied upon is made by mixing a good quality of graphite with raw linseed oil to the consistency of thin paint and then either rub or paint on with a brush, going over all bottom nipples and the mud drum, also the rear end of all the bottom tubes which may draw dampness from the bottom. When the combustion chamber is wet down for cleaning or when the boiler is being washed out, the water will soak up among the tubes, particularly the first and second rows. The blowoff pipes should receive the same treatment.

If the circulating nipples and the rear ends of the bottom tubes are kept dry and free from moisture they will not corrode externally or become thin except by the wear and tear caused by the passage over them of the escaping heated gases. It is seldom that tubes waste away on the inside, although this is sometimes the case where the feed water contains properties that cause pitting and corrosion. All baffle plates and arches should be examined carefully each time the boiler is washed out. The arches and furnace walls should be kept in perfect condition so as to insure a proper distribution of the heated gases. All manhead plates should be grooved and treated with a solution of graphite and oil; also all handhole plates and caps, as this will prevent corrosion and the wasting away of the plates in case of leakage.

If handhole plates leak and corrosion commences when a boiler is in operation, they should be treated with a thin solution of graphite and oil or kerosene oil. Kerosene is a most satisfactory remedy. From an oil can pour the kerosene on, commencing at the top of the header, and allow the oil to run down all over the headers, covering the handhole plates and all. This will also prevent corrosion and wasting away of the plates. If such precautions are taken, there is no good reason why water-tube boilers should not last twice as many years as they do; if they were properly constructed, erected and cared for, their life would be doubled.

The writer has in mind four water-tube boilers of 400 hp. capacity each, located in a large office building in a Western city. These boilers are used two at a time for 30 days; in the meantime they are thoroughly washed out and properly cared for. They have been in constant use for the past 10 years and they have never had one cent's worth of repairs outside of those necessary to the furnace walls and arches. These boilers are equipped with flat grates, hand fired in dutch ovens; the arch is extended back to about 6 ft. from the rear end of the tubes.

The evaporation under ordinary running conditions is between 8 and 9 lb. of water per pound of coal, and there has been no complaint from the smoke inspector. These boilers are cared for and managed along the lines suggested in this article. There is another plant, a duplicate of this one, where the management is just the reverse. The boilers are washed out once every four to eight months and new tubes are provided from time to time. When the inspector made an examination of the tubes recently, they were found almost half filled with scale. I doubt very much if the evaporation under ordinary circumstances would exceed 4 or 5 lb. of water per pound of coal. If the loss in fuel in operating this plant is compared with the one first mentioned, a good illustration will be had of what proper care and management mean in the economical operation of a plant.

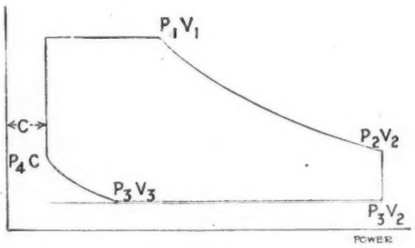
Altitude and Power Plant Economy

By Prof. A. G. Christie

In general, engine room economy is practically the same regardless of location. Boiler evaporative efficiency is independent of altitude, but an increase in grate area or draft is necessary.

Power plants in many of the new cities of the Western mountain states and of western Canada are located at altitudes greater than 2500 ft. above sea level. The writer was recently interested in the design of a plant to be about 3500 ft. above sea level and the question of the effect of altitude on the operation and efficiency of power-plant machinery, received considerable attention. Some engineers maintained that relatively as high vacuum or as good engine economy cannot be obtained at high elevation as at sea level. The following is taken from notes prepared at that time, in which the effect of altitude on both engine and boiler equipment is considered. It is hoped that this discussion will be of assistance to engineers who have to deal with plants similarly located.

In considering reciprocating engines at the two elevations, ideal indicator diagrams were analyzed both for condensing and noncondensing conditions with various degrees of clearance. Certain conditions were assumed and the mean effective pressure of each theoretical diagram was calculated as follows:



THEORETICAL INDICATOR DIAGRAM

The illustration represents a theoretical indicator diagram where P_1 is the admission pressure in the cylinder, V_1 the total volume at cutoff, P_2 the pressure at release, V_2 the volume swept over by the piston together with the clearance C. The back pressure is P_3, and V_3 is the total volume at the beginning of compression. Admission takes place at pressure P_1 and at the end of the stroke. The expansion and compression lines are assumed to be rectangular hyperbolas; that is, the product of absolute pressure and total volume at any point is a constant for each curve. If the pressures are measured in pounds per square inch, absolute, and the volumes in cubic feet, then the mean effective pressure of the diagram can be represented by the following formula, which is of the same form as the one given on page 931 (eighth edition), of Kent's "Mechanical Engineer's Pocket Book."

$$M.e.p. = \frac{P_1 V_1 \left(1 + hyp.\ log.\ \frac{V_2}{V_1}\right) - P_3\left[(V_2 - V_3) + V_3\ hyp.\ log.\ \frac{V_3}{C}\right] - P_1 C}{V_2 - C}$$

NONCONDENSING ENGINES

A small noncondensing engine, such as is often used with a throttling governor on auxiliaries, was considered first. Its operating conditions were assumed to be inlet pressure 100 lb., gage; cutoff, one-third, release at the end of the stroke, back pressure 1 lb. above the atmosphere, compression commencing at 20 per cent. of the stroke and the clearance volume 12 per cent. of the piston displacement. First consider such an engine in a plant at sea level with a barometer of 29.92 in. of mercury, equivalent to 14.7 lb. per sq.in., absolute, pressure. Representing the volume swept over by the piston by 100, and substituting in the equation at the bottom of the column

$$M.e.p. = \frac{114.7 \times 29\left(1 + hyp.\ log.\ \frac{104}{29}\right) - 15.7\left[(104 - 14) + 14\ hyp.\ log.\ \frac{14}{4}\right] - 114.7 \times 4}{104 - 4} = 54.27$$

At 3500 ft. elevation

$$M.e.p. = \frac{112.9 \times 29\left(1 + hyp.\ log.\ \frac{104}{29}\right) - 13.9\left[(104 - 14) + 14\ hyp.\ log.\ \frac{14}{4}\right] - 112.9 \times 4}{104 - 4} = 55.09$$

In this case the power of the same engine at high altitude exceeds that at sea level by

$$\frac{55.09}{54.27} = 1.0151$$

Several other cases were computed besides those given above. From these results the capacity or power of an engine running noncondensing evidently is practically unaffected by altitude as the increases shown above are of a very small order.

Suppose this engine moved to 3500 ft. above sea level where the barometer normally reads 26.25 in. of mercury or 12.9 lb. per sq.in. pressure. If the valve setting remains the same the only values that will change in the above equation will be the absolute pressure. Then

$$M.e.p. = \frac{114.7 \times 45.3\left(1 + hyp.\ log.\ \frac{112}{45.3}\right) - 15.7\left[(112 - 32) + 32\ hyp.\ log.\ \frac{32}{12}\right] - 114.7 \times 12}{112 - 12} = 67.48$$

$$M.e.p. = \frac{112.9 \times 45.3\left(1 + hyp.\ log.\ \frac{112}{45.3}\right) - 13.9\left[(112 - 32) + 32\ hyp.\ log.\ \frac{32}{12}\right] - 112.9 \times 12}{112 - 12} = 68.17$$

The power that can be developed in an engine cylinder is proportional to the mean effective pressure; hence, the power of this engine at the two altitudes is in the proportion of their relative mean effective pressures. Therefore, the power is increased by

$$\frac{68.17}{67.48} = 1.01\ per\ cent.$$

by moving the engine to the higher altitude.

To determine whether clearance and compression have any effect on the result, the same pressures were assumed as in the preceding case, while the clearance was made only 4 per cent. of the piston displacement, compression commencing at 10 per cent. and cutoff taking place at one-quarter stroke as in a small Corliss engine. At sea level

$$M.e.p. = \frac{114.7 \times 29\left(1 + hyp.\ log.\ \frac{104}{29}\right) - 15.7\left[(104 - 14) + 14\ hyp.\ log.\ \frac{14}{4}\right] - 114.7 \times 4}{104 - 4} = 54.27$$

At 3500 ft. elevation

$$M.e.p. = \frac{112.9 \times 29\left(1 + hyp.\ log.\ \frac{104}{29}\right) - 13.9\left[(104 - 14) + 14\ hyp.\ log.\ \frac{14}{4}\right] - 112.9 \times 4}{104 - 4} = 55.09$$

An engine's steam consumption cannot be accurately determined from an indicator diagram. The Mollier diagram is therefore used to compare the relative economy of an engine at the two elevations.

The efficiency of a steam engine or turbine is defined as the ratio of the percentage of heat turned into work in the actual engine per pound of steam to that which could be turned into work in an ideal engine without friction, radiation

or other losses and with the steam expanded adiabatically. This definition is fully explained in POWER, May 2, 1911, where a Mollier heat-entropy diagram is also shown. As far as present knowledge goes, the efficiency ratio of a given engine or turbine may reasonably be assumed to be constant at all elevations. Hence, the heat energy turned into work at the two altitudes will be proportional to the heat available from an adiabatic expansion from initial to final conditions in the two cases, the steam being assumed to be initially of 98 per cent. quality, which is common in practice.

By the above mentioned diagram, the heat available from adiabatic expansion from 114.7 lb. per sq.in. abs. and 98 per cent. quality to 15.7 lb. abs. exhaust pressure at sea-level conditions, amounts to 142.7 B.t.u. per lb. At 3500 ft. elevation 149.4 B.t.u. are available from a similar expansion from 112.9 lb. abs. and 98 per cent. quality to 13.9 lb. abs. at exhaust. Hence, each pound of steam at 3500 ft. elevation will produce

$$\frac{194.4}{142.7} = 1.047 \text{ lb.}$$

more work than at sea level, or, conversely, at this altitude the engine will only require 95.5 per cent. of the steam used at sea level. Apparently, therefore, a noncondensing engine or turbine is more economical at higher altitudes than at sea level.

CONDENSING ENGINES

Assume a compound Corliss condensing engine with 4-to-1 ratio of cylinders and cutoff at one-quarter stroke in the high-pressure cylinder. Let the steam pressure be 150 lb. gage and the clearance in each cylinder 4 per cent. Let compression commence 10 per cent. from the end of the stroke in each cylinder, and assume an exhaust pressure of 2 lb. abs. As the receiver losses and diagram factors will remain practically the same in both cases, they will not be considered and the mean effective pressure will be calculated for the whole of the combined diagram. At sea level conditions

$$M.e.p. = \frac{164.7 \times 10.25\left(1 + hyp.\ log.\ \frac{104}{10.25}\right) - 2\left[(104 - 14) + 14\ hyp.\ log.\ \frac{14}{4}\right] - 164.7 \times 4}{104 - 4} = 47.38$$

at 3500 ft. altitude

$$M.e.p. = \frac{162.9 \times 10.25\left(1 + hyp.\ log.\ \frac{104}{10.25}\right) - 2\left[(104 - 14) + 14\ hyp.\ log.\ \frac{14}{4}\right] - 162.9 \times 4}{104 - 4} = 46.77$$

The heat available from adiabatic expansion under sea-level conditions with 98 per cent. initial quality is 283.4 B.t.u. per lb. and 283 B.t.u. in the second case, so that the steam consumption of an engine under such conditions evidently is practically unaffected by altitude. On the other hand, the power of the engine, as shown by the mean effective pressure at 3500 ft. elevation is only 0.9885 of its power at sea level with the same valve setting. Hence, power decreases with altitude while the steam consumption may be said to remain constant. With a steam turbine, the exhaust pressure would probably have been maintained at 1 lb. abs., and superheat would also likely be used. Assuming 100 deg. F. superheat at admission, the heat available at sea level would be 343 B.t.u. per lb. and at 3500 ft. 342.3 B.t.u., which again shows that the steam consumption is practically unaffected by changes in altitude. These deductions are important to purchasers of machinery to be located at high altitudes for manufacturers should give practically as good guarantees on their turbines under these conditions as at sea level.

At the higher altitudes the air will be less dense and hence the loss due to air friction and windage in the flywheel and generator of an engine or in the rotor of a turbo-alternator will be less than at sea level. This increase in efficiency will be relatively small.

With inclosed turbo-alternators, a greater volume of air will have to be forced through the rotor to produce the same cooling effect as at sea level through the work done in forcing such air through will remain practically the same.

With regard to the heating of generator windings at high altitudes, the Standard Rules of the American Institute of Electrical Engineers for testing electrical machinery, as given in Foster's "Electrical Engineers' Pocket Book," state that for every 0.39 in. difference in barometric pressure from 29.92 in., the observed temperature rise is to be reduced 1 per cent. to compare with standard guaranteed temperature rises. For the case under consideration at 3500-ft. elevation a correction of 9.54 per cent. would be made. This rule fits all cases very well, whether the generator has open ventilation or forced ventilation as in a turbo-alternator. This increased temperature rise indicates that the heat losses in a generator will increase with altitude. Hence the generator efficiency will decrease slightly at high elevations.

ENGINE ROOM AUXILIARIES

Some engineers believe that as good a relative vacuum cannot be maintained at a high altitude as at sea level. The factors affecting the vacuum in any condenser, if sufficient cooling water is available, are chiefly air entrained in the boiler-feed water, air entrained in cooling water, if a barometric or jet type of condenser is used, and air leakage.

The amount of air entrained in the boiler-feed water, and thus carried through the engine to the condenser, depends largely on the temperature of the feed water. As the temperature of this feed water approaches the boiling point at atmospheric pressure, it will contain less air if an open heater is used. Apparently, therefore, this is a question of temperature alone and hence is not affected by altitude.

A bottle filled with water saturated with air at sea level and then uncorked at a high altitude will behave similarly as a freshly opened ginger ale bottle—bubbles would appear as soon as the pressure is released. Hence at a high altitude water will not carry as great a weight of air when thoroughly saturated and at the same temperature, as at sea level. The amount of entrained air in conditions between absolutely pure water and saturated water will vary with local conditions, but generally less entrained air will be in the cooling water at high altitudes than at sea level.

Regarding air leakage, the hole through which the leak occurs, whether a pin hole or a gasket leak, may be considered as a small orifice. Air practically follows the laws of perfect gases. Hence, if the absolute pressure on the inside of the vessel is less than 0.58 of the outside pressure, the quantity of air flowing in will remain the same. The absolute pressure in the condenser is usually less than 58 per cent. of the atmospheric pressure regardless of the altitude. Therefore, with the same conditions as regards air-leakage holes, the amount of air leaking into the condenser will be constant regardless of altitude.

The foregoing shows that probably the air pump would have less air to handle at the higher altitudes than at sea level, due to the smaller quantity of entrained air in cooling and feed waters, and hence with the same sized air pump a higher vacuum might be carried with jet or barometric condensers.

The work done by the air pump will be less at the high altitude. For example, at 3500 ft. altitude and 1 lb. absolute condenser pressure the air pump has to compress less air from 1 lb. abs. to 12.9 lb. abs. pressure, while at sea level it compresses more air from 1 lb. abs. to 14.7 lb. abs.

The work required to drive the circulating pumps with either barometric or surface condensers under similar conditions of temperature and vacuum will be

practically the same regardless of altitude. The wet-air pump of a jet condenser has to lift the water against a smaller absolute pressure at the high altitude than at sea level, and hence does less work.

Therefore, relatively as high vacuum can be obtained at the high altitude as at sea level, and this can be produced at a lower operating cost. Consequently, striving for the highest possible vacuum at the high altitudes when steam turbines are used is advisable.

A thermos bottle would lead one to conclude that, in general, radiation losses from boilers, piping, etc., would be less at the lower atmospheric pressure of elevated locations. While such a decrease in air density no doubt has some slight effect, experiments along a similar line failed to show sufficient decrease in radiation loss to warrant consideration. Hence, in a commercial sense, radiation losses are independent of elevation.

Summarizing briefly, all noncondensing engines as exciters, air pumps, steam-driven circulating pumps and boiler-feed pumps will show better steam economy at the higher altitudes, the condensing apparatus will also take less power while the main engines will consume very slightly more steam. In general, then, as far as the engine room is concerned, the steam economy probably will be practically the same regardless of location.

Effect of Altitude on Boiler Plant

Under usual operating conditions 20 lb. of air is required per pound of coal burned in the boiler furnaces. If this air enters the furnace at 60 deg. F., 1 lb. of air will occupy a volume of 13.07 cu.ft. at sea level, while at a 3500-ft. elevation it will occupy 14.89 cu.ft., at the same temperature. That is, the furnace at the high elevation must be supplied with 14.5 per cent. greater volume of air than a furnace at the lower elevation. Then, either the area of the grate must be increased 14.5 per cent. with the draft remaining constant, or the draft must be increased sufficiently to provide for this extra volume of air. Therefore, with the same furnace economy and burning the same amount of coal per hour, furnaces at the higher altitudes evidently must either have increased grate area or more draft than similar furnaces at sea level. Neglect to consider this requirement has several times resulted in limiting the capacity of boilers to loads much lower than they would be capable of carrying under more favorable conditions.

It has been suggested that such an increase in the volume of air would promote better combustion in the furnace as each particle of combustible would come in contact with a greater volume of air. The proportion of oxygen and nitrogen remains practically the same at all points on the earth's surface, so that each proportional volume of air has a combining power with carbon exactly in proportion to its weight. In the ordinary boiler furnace, apparently large volumes of air pass through the furnace without even coming in contact with the carbon. It would therefore seem probable that if the draft were increased at the higher altitude, the furnace area remaining constant, improved combustion would result from this increased volume of air striking the burning carbon at increased velocities over that occurring in a similar furnace at sea level.

If the absolute pressure is the same altitude has no effect on a steam boiler, for the heat required to produce steam at a given absolute pressure from feed water at a constant temperature, remains the same. This is shown by any steam table. If, however, the pressure above atmosphere, or, the gage pressure, is the same in both cases, there is a slight difference which will be illustrated by an example.

Suppose the boiler gage pressure to be 150 lb. and the feed-water temperature 182 deg. F. both in a boiler at sea level and in one at 3500 ft. elevation. According to the steam tables the boiler at sea level and at 164.7 lb. absolute pressure requires

$$1194.97 - (182 - 32) = 1044.97 \text{ } B.t.u.$$

to evaporate 1 lb. of steam from feed water at 182 deg. F. A similar boiler at 3500 ft. elevation or 162.9 lb. absolute pressure would require

$$1194.79 - (182 - 32) = 1044.79 \text{ } B.t.u.$$

to evaporate 1 lb. of steam from the same feed-water temperature. Hence, a difference of elevation of 3500 ft. decreases the heat required to evaporate 1 lb. of steam under the given conditions by 0.18 B.t.u. or 0.018 per cent. approximately. This fraction is so small that for practical purposes it is negligible. Therefore, with regard to evaporative efficiency, in commercial operation, altitude has no effect on steam boilers.

That a greater volume of air will be required in the furnace at the higher altitude has already been shown, so that a greater volume of furnace gases must pass into the boiler tubes or flues from the furnace itself. Boilers are usually built from standard designs so that the areas for the passage of gases are the same in all similar boilers. These areas are usually figured for sea-level conditions. Evidently, therefore, since a greater volume of gas passes through the same area, the gases at the higher altitude must pass through the flues or tubes at a much higher velocity than in a similar boiler at sea level, if the same amount of coal is burned with the same excess of air in both cases. Experiments recently made in England show that increased velocity of the gases through a boiler results in increased boiler efficiency. That the boiler at the higher altitude should show better efficiency than the one at sea level would therefore be expected, though it is hard to predict the amount of this increase from the available information on the subject.

With the higher flue-gas velocities, one can expect higher friction losses in the boiler flues or tubes. Hence, at the high altitude the draft required to overcome these friction and eddy losses will exceed that at sea level irrespective of the area of the grate itself. When the grate area remains constant, the draft at high altitudes will be considerably greater than at sea level, as already pointed out. Therefore, if an induced-draft equipment is provided, the fans, if of the same size in both plants, must be run faster at the high altitude. This would result in a very decided increase in the power required. A larger fan provided in this plant would also require more power to drive it than the one at sea level. Evidently, then, an induced-draft plant requires more power in a plant at high altitudes than in one at sea level.

The height of a chimney required under any given condition is determined by the formula (from Gebhardt's "Steam Power Plant Engineering").

$$H = \frac{D_1}{0.52 P_2 \left(\frac{1}{T_2} - \frac{1}{T_1}\right)}$$

where

H = Height of chimney in feet;
D_1 = Required draft in inches of water;
P_2 = Observed barometric pressure;
T_2 = Absolute temperature of outside air;
T_1 = Absolute temperature of the flue gases in the chimney.

This shows that the height of a chimney varies inversely with the barometric pressure, if the draft remains the same, which means the higher the altitude the higher the chimney required. But, as has been shown, friction losses in the boiler increase at the high altitude and hence greater draft will be required, resulting in a still higher chimney. If the grate area is not increased, still more draft will be necessary, and hence more height must be added.

Kent's formula for area of chimneys gives effective area in square feet.

$$E = \frac{0.3 \, hp.}{\sqrt{H}}$$

where $hp.$ = Horsepower of boilers served and H = Height of stack in feet.

The higher the chimney, then, the less is the effective area required. But the effective area varies inversely as the square root of the height, so the height will increase relatively faster than the diameter decreases. A chimney's cost is usually determined largely by the height. Hence, at a high altitude a chimney must be higher with somewhat smaller diameter, but with better foundation, and will cost more than one for similar boiler power at sea level.

Designing Brick and Steel Chimneys

By Everard Brown

Determining height and area of stack, wind pressure and stability, methods of construction and conditions governing size of foundation.

In designing chimneys the height is generally decided upon first. This involves consideration of the height of surrounding buildings or hills, the length of all horizontal flues necessary, the character of fuel to be used, etc., as well as any local laws or ordinances that may apply. The chimney diameter may then be calculated, based upon the selected height and the boiler horsepower or the amount of fuel to be consumed.

The minimum height necessary, however, will be governed to a great extent by the fuel—wood usually requiring the least and fine anthracite coal the greatest—the character of the installation and the number of furnaces served. Obviously the smaller and more round-about the flues, the higher must be the stack; likewise, a single furnace will require less height than several discharging into a common flue. On account of these variables, it is practically impossible to evolve a general formula applicable to all cases. Except in rare instances, extremely tall chimneys are not necessary and any increased efficiency gained by them hardly ever justifies their greater cost. The tendency is to build two or more smaller ones rather than one high one, which can very often be done at less cost.

A chimney should be proportioned to give sufficient draft for the boiler to develop much more than its commercially rated power, or, according to Kent, to bring about the combustion of 5 lb. of fuel per rated boiler horsepower per hour. By assuming such a liberal rate of combustion, the stack will certainly be large enough. Based on this assumption Kent gives the following equation:

$$E = \frac{0.3\ Hp.}{\sqrt{H}} = A - 0.6\sqrt{A}$$

where:

E = Effective sectional flue area of stack in square feet;
$Hp.$ = Horsepower of boilers;
H = Height of chimney;
A = Actual sectional flue area of stack in square feet.

The actual is not the effective sectional area because friction retards the ascending gases. The annular stratum of gas retarded by the wall in reality diminishes the flue area.

Molesworth gives:

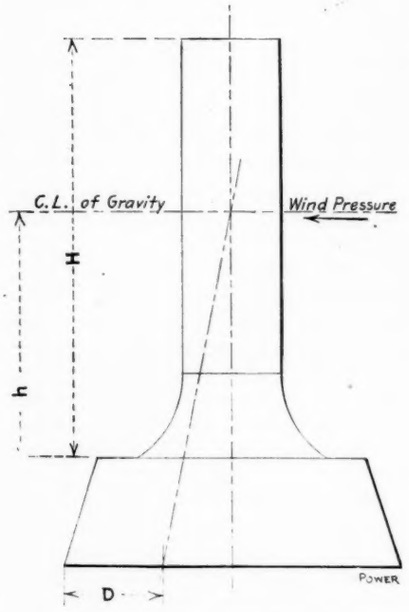
$$A = \frac{Hp.}{1.28\sqrt{H}} = \frac{C}{12\sqrt{H}}$$

where

C = Pounds of coal consumed per hour.

From this equation, H may be found if the sectional area is known or assumed, or by the same process the horsepower may be found for which an existing stack is sufficient. Sometimes windcaps are used to increase the stack capacity, but this is only rarely advisable because of their dangerous position.

Stability and wind resistance are next to be considered. Wind pressure is usually assumed to be horizontal and uniformly intense at all levels. Where a chimney is shielded by buildings, only the part actually exposed need be considered. Generally a maximum pressure is assumed of 50 lb. per square foot,



DETERMINING STABILITY OF CHIMNEY

although this would require a wind velocity seldom reached. A simple rule for calculating wind pressure is:

The square of the velocity in miles per hour divided by 200 *equals the pressure in pounds per square foot.*

The total pressure against a round chimney is about one-half that against the diametral plane of that chimney. It is therefore the product of the pressure per square foot by one-half the total projected area of the exposed part. Similarly the total pressure may be calculated for chimneys of almost any form.

The resultant of this pressure may be assumed to act horizontally through the center of gravity of the exposed part. The stability of a wholly exposed chimney, therefore, is determined by finding the moment of this pressure acting through the center of gravity. From the accompanying illustration this moment is

$$h \times P_1 = H \times P \times \frac{A}{2}$$

where

h = Height of the center of gravity above the top of the foundation;
P_1 = Total pressure exerted;
H = Total height of the stack;
P = Pressure in pounds per square foot;
A = Total vertical sectional area.

For chimneys with vertical axes, the moment of stability is the same in every direction, but few chimneys have exactly vertical axes; therefore, the least moment of stability must be considered as that which opposes the lateral pressure in the direction in which it leans. The stability at any height of the chimney, for example at a certain bed-joint of a brick structure, may be established in the same way by simply considering that part of the chimney above the point at which the stability is to be determined. Opinions differ regarding the strength of brickwork, which, together with the varying qualities of materials and workmanship, makes it exceedingly difficult to accurately calculate the power of resistance. Consequently a large factor of safety is always advisable.

Fundamentally, a chimney's stability depends upon the weight of its outer shell and the diameter or width of its base and, while the cohesion of the mortar in a brick stack may add something to its strength it is too uncertain to be relied upon. The effect of the two forces—the weight of the chimney and the pressure of the wind—is to shift the center of pressure at the base from the axis toward one side, extent depending upon the relative magnitude of the two forces. A comparatively safe rule to follow is to make the base diameter not less than one-tenth of the height. This rule has been approved by many years of actual practice.

STEEL CHIMNEYS

Probably the cheapest chimney is a straight steel shell guyed with rods or wires. These are secured to the shell, usually by an encircling band, and should be anchored to incline at an angle of not more than 45 deg. with the ground. Where other structures are convenient the guys may be fastened to them. This type of stack is rapidly being supplanted by the so called self-sustaining stack. It can be lined or not. Ordinarily when

working to full capacity, the high velocity of the ascending gases allows little heat to be lost by radiation even through an inclined shell. If the gas temperature is high, a lining is desirable at least part way up. Firebrick is sometimes used, but probably more often hard-burned red brick that can withstand temperatures up to 750 deg. without injury. This lining should never be less than 4½ in. thick at the top, whether it goes up only part or the full height of the stack, and should be thicker toward the bottom by 4½ in. for about every 35 or 40 ft. Not less than 1 in. should be allowed between the shell and the lining with no rigid bond between them, thus making both self-sustaining and allowing for expansion and contraction.

For ease of construction and economy, uniform diameter of the shell for all sections is preferable; the bottom of each section should lap outside the top of the next section below, except for brick-lined structures, when the reverse is often preferred. The rivets in horizontal joints should be double staggered, thus giving more lap for the sheets and, consequently, more stiffness to the shell. Single-riveted joints usually answer for vertical seams. If possible, each section should be made of a single plate, to avoid the extra riveting. In the flaring base both the vertical and the horizontal seams should be double staggered. The cone-shaped base is naturally the strongest, but bases curved like a bell are quite common, and may be equally strong if thicker material is used. Better appearance is the only advantage of the curved base.

Occasionally instead of either cone- or bell-shaped bases, the shell is made perfectly straight down to the foundation and anchored to it by steel braces or cables with turnbuckles. The latter are usually fastened to the stack at a height of about three diameters of the flue and have approximately the same slope as the conical base. This type of construction is hardly advisable.

Caps or other ornaments at or near the top of a steel chimney are no use and a poor investment, but if desired should be of copper to be quite light and rust proof, and firmly attached to prevent risk of being blown down, causing damage or human injury.

The chief advantages of the self-sustaining steel chimney over all others are less space occupied above the ground, and easier and more rapid construction and erection; and over brick chimneys in particular, lighter weight often saving pile driving in low load-sustaining soils; less area presented to the wind; elimination of air infiltration and consequent checking of the draft; greater strength and safety, and smaller cost. Places and conditions where steel shell construction would not be at all practicable nor economical are, for example, along the sea coast or where acid fumes are present in the atmosphere.

Brick Chimneys

The most common brick chimney is the radial type. While initially costing somewhat more usually than a steel structure, it does not have the latter's expense for painting, replacements, etc. A brick chimney is usually made up in two shafts, the shell or outer casing and the lining or core. There is no bond between these rigid enough to retard movement due to expansion caused by the hot gases. Generally the core is run up to within a few feet of the top and then battered to connect with the shell. The outer casing diminishes in thickness, section by section toward the top. The joints between sections and at the base, called bed-joints, are less stable than intermediate ones and always must be given primary consideration. A failing chimney nearly always begins by opening one of these bed-joints on the windward side and cracking along both sides diagonally downward, tending to separate the structure into two parts: an upper leeward and a lower windward. Therefore, to withstand extreme intensity of wind pressure, the vertical pressure should be sufficient to counteract and prevent tendency for the joint to open.

The lengths of each section vary according to the height of the chimney and the quality and kind of brick used. For ordinary red brick the lengths may be from 25 to 30 ft. with an outside batter of 1 to 30 or 35, or they may increase gradually from the lowest division to the highest. Using bricks of greater crushing strength, these divisions may be made longer. The wall thickness and the batter must conform to the various lengths to insure stability. The former, giving weight to the structure, and the latter, presenting a sort of lever arm, combine to resist wind pressure. Since a swaying motion is usually set up by the wind, it is never desirable to connect the chimney rigidly with any other structure.

Naturally, much depends upon the workmanship and material; both should be of the best. Important, also, is not to erect too rapidly for the mortar to properly set and the foundation should have plenty of time to harden and settle. In several ways the brickwork may be bonded satisfactorily. The most common bond is probably one row of headers for every five rows of stretchers, although alternate rows of headers and stretchers, or two of the latter for one of the former, are also used. Often in large chimneys an iron or steel hoop or tee iron is laid within the brick wall every few feet. This is good, especially where the wall thickness exceeds, say 10 to 12 in.

Concrete Chimneys

Reinforced-concrete chimneys are least common and were introduced only about 10 years ago. Information regarding them is meager and they have hardly had time to demonstrate their durability. Ordinarily, they have inner and outer shells with an annular air space between. The shell thickness depends upon height, diameter, etc. The steel reinforcement is usually vertical bars with ends overlapping, spaced according to the size of the chimney. These bars are encircled by steel rings and are extended down into into the foundation to insure a good anchorage. They are calculated to resist all pressure by the wind and any tensile strength in the concrete is usually considered an addition to the safety factor.

Foundations

Logically, the first detail is the foundation and in its design must be considered any local building laws that apply. These laws usually specify the loads allowed in tons per square foot of area for ground of various kinds. Where the soil is very loose or sandy, piling is often necessary, but, regardless of expense, a solid, amply large foundation is imperative. Brick, stone or concrete may be used either individually or collectively. Concrete is commonly used for the subfoundation with brick or large stone above. The area of the foundation should be sufficient to prevent unequal settling. It will receive greater pressure from above on the leeward side to the prevailing wind, resulting in a canting of the chimney if the foundation is not broad and substantial. The foundation should be such that wind pressure against the chimney will not unduly increase the load on it at any point.

To determine the stability of the chimney in relation to its foundation: first, find the greatest total wind pressure to which the chimney might be subjected and its moment about an axis in the plane of the base of the foundation; second, the total weight of the chimney, including its lining and also that of the foundation itself. The former divided by the latter will give the distance from the outer edge of the foundation, or the lever arm of the combined weight of the chimney and foundation producing equality of moments. If this distance D is one-half the foundation diameter at the base, the chimney may be considered as just stable with no margin of safety; as it is lessened the factor of stability is, of course, increased.

This distance should never be more than one-third that across the base of the foundation even with the best construction and anchorage. In addition some strength will be obtained from the ground surrounding the foundation, but it will be too variable to be counted on. Evidently, the greater the combined weight of the foundation and chimney, the more stable will the whole structure be made.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Three Wire Direct Current Generators

BY A. M. BENNETT

Three-wire generators are designed to furnish two voltages, generally 250 and 125, the former between the two outside wires and the latter between each outside wire and the middle wire.

The chief advantage of a three-wire system over the ordinary two-wire installation is that of economy in distribution. In a two-wire system, with a given load and a given percentage of voltage drop, the distribution at 250 volts requires only one-quarter the weight of copper required for distribution at 125 volts. A third or neutral wire in the three-wire system will modify this proportion of copper, the final saving depending on the size of the neutral. In well designed systems, the maximum unbalanced current carried by the neutral will be about 25 per cent. of the full load. Therefore, the size of the neutral need not be larger than 25 per cent. of the capacity of the outside mains. The weight of copper in this case would be $\frac{9}{32}$ of that used in distributing the same power by a two-wire system.

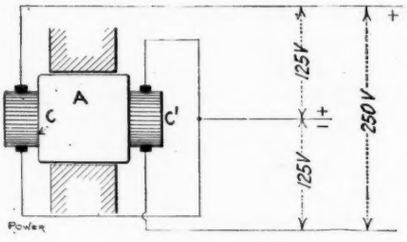


FIG. 1. HOPKINSON THREE-WIRE GENERATOR

There is also the advantage in the three-wire system of having at hand two voltages; the higher can be used for power purposes and the lower for lighting. Motors can be designed to work equally as well on the higher voltage as on the lower, and in the larger sizes their cost will be less. They are connected directly across the two outside wires and do not affect the unbalanced load. The lower voltage, namely, 110 or 125 volts, has been found most satisfactory for lighting, the lamps being connected between the neutral and each outside wire; the total load of this nature is divided so that the loads on the two sides are at all times as nearly balanced as possible. The difference between these loads constitutes the unbalanced load of the system, which is carried by the neutral wire.

As explained, the difference between the two-wire and the three-wire generator lies in the ability of the latter to furnish an extra voltage at one-half that which obtains between the regular leads of the generator. This must be done in all cases with close voltage regulation between the two sides of the system when carrying an unbalanced load. Only when the two sides are equally loaded is the voltage between the neutral and the outside wires the same. As one side becomes more heavily loaded and the unbalance increases, the voltage drops below that on the side with lighter load until at the point of maximum unbalance, for which the generator is designed, there will exist the maximum difference of voltage between the two sides of the system. Any great difference that may exist will have a detrimental effect on the life and efficiency of the lamps.

Some of the best known three-wire generators now in use, and which most successfully accomplish the desired results are included in the following:

HOPKINSON THREE-WIRE GENERATOR

This is represented in Fig. 1, and is among the oldest types of three-wire machines. The armature *A* has two similar windings, and two commutators *C* and *C'*. Each winding is designed for one-half the total output of the generator at 125 volts. The two sides of the machine are connected in series, and their voltages are added to produce 250 volts on the outside leads. The neutral is taken off at the point of connection between the two windings. There exists, therefore, 125 volts between the neutral and each outside main. The result is similar to that obtained by the Edison three-wire system, where two 125-volt generators were run in series, with a neutral taken off at the point of connection between the two machines.

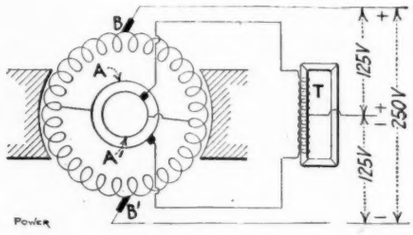


FIG. 2. DOBROWOLSKY THREE-WIRE GENERATOR WITH SINGLE-PHASE BALANCING COIL

In the Hopkinson generator, however, since the two armature windings are influenced by the same field,

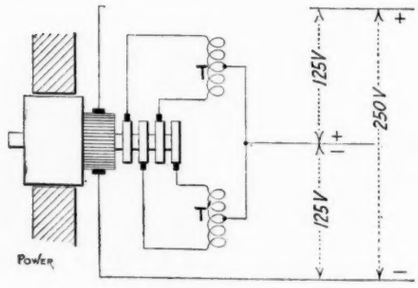


FIG. 3. WITH TWO-PHASE BALANCING COIL

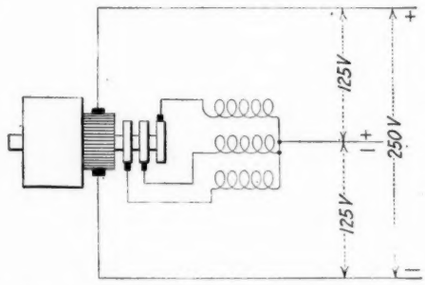


FIG. 4. WITH THREE-PHASE BALANCING COIL

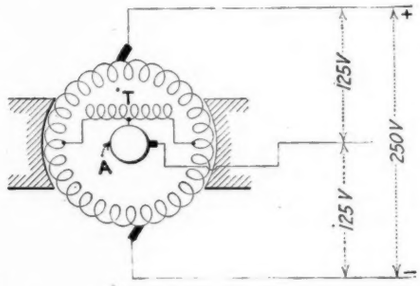


FIG. 5. WITH SELF-CONTAINED BALANCING COIL

MODIFICATIONS OF THE DOBROWOLSKY THREE-WIRE GENERATOR

there is no means of compensating for the difference in voltage between the two sides with an unbalanced load, except by shifting the brushes. This would hardly be practical, as it would have to be done on each change of load from one side of the system to the other.

Dobrowolsky Three-wire Generator

The majority of straight three-wire generators running at present are of this type or one of its modifications. It may be regarded as an adapted two-wire generator, in that it consists of such a machine working in connection with an ordinary auto-transformer, or balancing coil. This transformer has a single winding on a laminated core. Its outer terminals are connected to brushes working on two slip rings mounted on the armature shaft, these rings in turn being tapped on to the armature winding at two points of opposite potential, forming a single-phase connection to the armature. The neutral

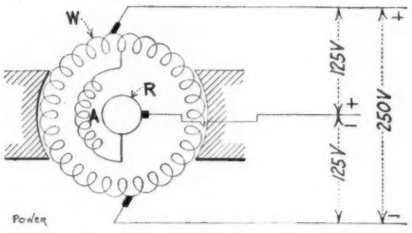


Fig. 6. Osana Three-wire Generator with Single-phase Balancing Coil

wire is taken off at the middle of the transformer. This is shown diagrammatically in Fig. 2, where T is the auto-transformer, A and A' the slip rings connected to the armature winding, and B and B' the main brushes.

As the coils of the transformer are wound on a laminated core, they have a high inductance. They are also subjected to an alternating potential from the armature, and by reason of the counter electromotive force generated in them, do not short-circuit the armature. The middle point of the transformer is always at a potential halfway between that of its two terminals; since the potential between the neutral wire and either outside lead of the generator will be one-half that between the two outside wires. The unbalanced current on the system may flow in either direction in the neutral, depending on which side is more heavily loaded. In any case, the neutral current is divided in the balancing coil. This current meets with no opposition in the coil other than that of ohmic resistance, and as it has opposite directions around the core, its magnetizing effect on the latter is zero. There is, therefore, no interference from this source with the choking effect of the balancing coil on the alternating current to which it is subjected.

In practice the auto-transformer is generally built in compact form and mounted in a cast-iron case filled with oil. It can be placed in any convenient location, the leads from it being taken to the slip rings on the generator.

A modification of the Dobrowolsky system is shown in Fig. 3. Two balancing coils, T and T', and four slip rings are employed, and the connection of the latter to the armature is two-phase. The middle points of the two balancing coils are connected and the neutral is taken off from this common connection. By this construction the path of the neutral current through the armature winding is shortened somewhat, and there is a more uniform action of the coils, with less tendency to cause a flickering of lamps on the system in the case of slow-speed machines, where the alternations are low, than with the single-phase connection. There is the disadvantage, however, of two added slip rings with their brushes, and extra leads to the machine; also the machine is longer.

Still another modification of this system, claimed to be particularly free from unsteadiness of the neutral current is shown in Fig. 4. Here three balancing coils are employed connected in "star," and three slip rings, whose connection to the armature winding is three-phase. The neutral wire is taken off at the point of common connection of the three coils.

There is a fourth modification of this type in which the balancing coil is mounted with the armature on the shaft. The outer terminals of the coil can be connected directly to the proper points on the armature winding. The neutral is taken from the middle point of the balancing coil as before, and is carried to a single slip ring, from which connection is made to the outside circuit. This is shown in Fig. 5, where T is the auto-transformer with its middle point connected to the slip ring A, and its terminals connected to the armature winding. This construction gives a very compact arrangement and the auto-transformer gets the benefit of ventilation, due to rotating with the armature. The balancing coils in this construction can be connected either single-, two- or three-phase, with the same advantages as where these coils are separate from the machine, but in any case only one slip ring is required.

The Dobrowolsky generator and its modifications have the disadvantage that it is not possible without extra mechanism, to regulate the voltage on the two sides of the system. At a given unbalanced load, the difference in voltage is fixed and is determined when the machine is designed.

Osana Three-wire Generator

In this generator, which has now come into extensive use in this country the unbalanced current is taken care of by an auxiliary winding laid in the bottom of the armature slots, beneath the main winding. This is represented diagrammatically in Fig. 6, where W is the main winding, A the auxiliary winding, and R the slip ring. The auxiliary winding is tapped to the main winding, and between the point of connection to the armature and the slip ring, has just one-half the number of turns that are in the main winding, in series between any two sets of brushes. In the position shown it is evident that the coil is at its maximum generating point, and in this position, the voltage generated by it, and existing between the neutral and outside mains, will be just one-half that between the outside mains. At 90 deg. to this position it is at its minimum or zero generating point, and the neutral voltage is generated in the main armature winding itself, the auxiliary winding serving merely as a conductor for the neutral current. Between these two positions the neutral voltage is the sum of the voltages generated in the auxiliary winding, and in that part of the main winding between the point where the former is tapped to the latter, and one of the brushes. This will always be one-half the voltage between the outside mains of the generator.

The single-phase winding shown in Fig. 6 concentrated under a single pole, has not given a steady voltage, and in practice, a polyphase winding (Fig. 7) connected at three or more points to the armature is used. This winding is distributed in such a manner, that the average flux cut by it is at all times practically uniform.

This type of machine is particularly efficient, inasmuch as core losses and excitation losses present in the Dobrowolsky type are removed. It has the disadvantage, however, of requiring an armature of larger diameter, on account of the extra room needed in the slots for the auxiliary winding. The same fault that characterizes other three-wire generators,

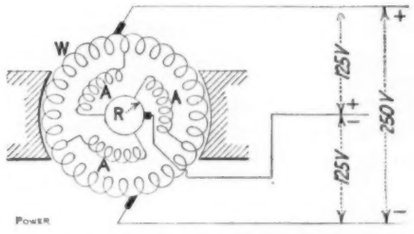


Fig. 7. Osana Three-wire Generator with Balancing Coils

namely, lack of means for regulating the voltage on the two sides of the system, is also present in this generator.

Dettmar and Rothert Three-wire Generator

While the Dobrowolsky and Osana generators bear some relation to each other in the manner in which the neutral voltage is generated, the Dettmar & Rothert machine is a distinct departure from either of these types. In this generator, shown diagrammatically in Fig. 8, a two-pole armature is rotated in a four-pole field. Poles of like polarity are placed next to each other forming in reality a two-pole field, with a gap in the center

of each pole. Opposite this gap a third brush is placed on the commutator, and the neutral wire is led off from this point. The effect of armature reaction, unlike that in the machine with normal poles, is to strengthen opposite north and south poles, and to weaken the remaining two. Like poles on each side of the neutral connection thus have unlike strength, with the result that the voltage on one side of the system is higher than on the other, even on a balanced load. This is

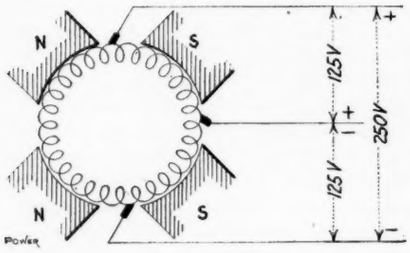


FIG. 8. DETTMAR & ROTHERT THREE-WIRE GENERATOR

compensated for by the manner of connecting the compound winding on the fields, that on the strengthened poles being connected differentially, and on the weakened poles cumulatively. This method of winding regulates the voltage automatically on the two sides of the system, raising it on one side and lowering it on the other, depending on which side has the greater load. The no-load voltage on the two sides of a system fed by such a machine can also be regulated by dividing the shunt field windings into two circuits and placing a rheostat in each.

All three-wire generators, in order to regulate the voltage between the outside mains, have their series fields divided into two circuits, alternate poles being connected in the positive lead from the generator, and the remaining poles in the negative. They can be operated in parallel as satisfactorily as two-wire machines, but two equalizing connections must be provided, one for each series field circuit. Inasmuch as these equalizers are each connected to one terminal of the machine, they have the full potential of the generator across them. Adjacent series fields are also subjected to this same condition. Care must be taken, therefore, that there is no chance for a short-circuit between any of these parts.

Three-wire generators can be operated in parallel with two-wire generators without trouble, but the series fields of the two-wire machine must be arranged in two circuits, and two equalizers provided. In this case the three-wire machine handles the unbalanced load as if it were operating alone. In any installation, therefore, it is necessary to operate only one three-wire generator, provided that it is designed to take care of the full unbalanced load of the system.

In Fig. 9 are shown the connections of a three-wire generator to its switchboard. There are six cables to the board, each carrying the full armature current, and one for the neutral, which is smaller than the others. This number of cables is necessary, for in order to fully protect the generator the full armature current must flow through the circuit-breaker coils. This means that the armature leads must go direct from the machine to the switchboard and back, before passing through the series fields. From the series fields the leads are taken to the busbars on the board. The equalizing connection is made at the switchboard instead of at the machine, as is the case with two-wire generators. The ammeter shunts are also connected in the leads from the armature ahead of the equalizer. A two-pole circuit-breaker is necessary to fully disconnect the generator from the line, and its two poles must be interlocked, so that both go out together. Two double-pole and a single-pole switch are also necessary, placed as shown. This allows the series field to be thrown into circuit before starting the machine. The circuit-breaker must be open till the voltage is built up to that of the line, otherwise the armature of the incoming machine would

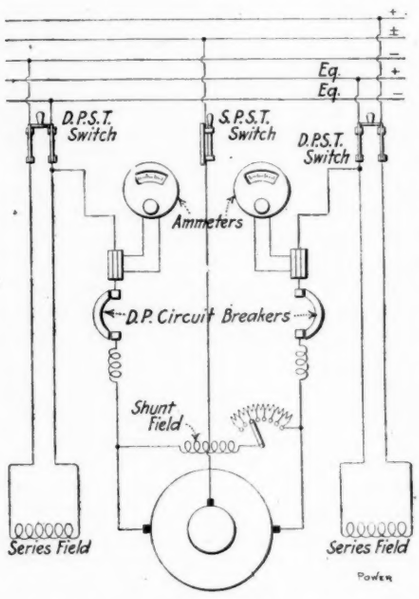


FIG. 9. SWITCHBOARD CONNECTIONS FOR THREE-WIRE GENERATOR

act as a dead short-circuit to the rest of the machines on the line. Two ammeters are employed, one to read the current on each side of the system, the difference between the two readings being the unbalanced current. It is not necessary to have an ammeter in the neutral line.

It is not good practice to insert a circuit breaker in the neutral unless it is also designed to open the main lines; for if the neutral alone were opened, the neutral load would be thrown across the outside mains of the system. For the same reason fuses should not be used in the neutral line.

There is another method of connecting the generator and switchboard, in which two of the leads shown in Fig. 9 are dispensed with. In this case the ammeter shunts are mounted on the frame of the generator, in circuit between the armature leads and the series fields, and the circuit-breakers are placed in the outside lines. It is necessary in this case that the circuit-breaker open the equalizer circuit at the same time that the main circuits are opened, otherwise the machine is not disconnected from the line.

CORRESPONDENCE

Reversed Polarity

A letter by James Mitchell in the July 2 issue headed "Reversed Polarity," recalls a personal experience with reversed machines. A few years ago, I was in charge of an electric plant consisting of five generators, belted to three engines and supplying power to motors and lights in a machine shop. The generators were compound wound, each being connected to the busbars by a three-pole switch with the equalizer busbar mounted on the same board.

One day as we were starting up after the noon hour the engines were up to speed and four of the generators were connected to busbars, the fifth was just being thrown in when there was a sudden squealing of belts, and the breakers opened on the machines already on the line. It was at once evident that the machines were reversed in polarity, but we did not know how many were in this condition. We tried each machine and found one that was correct, and three of them reversed. As it was getting near 1 o'clock we opened the largest feeder switches so that the motors in the shop could not be started until everything was all right.

We threw in the good machine on the busbars and proceeded to correct the polarity of the others in the following manner: We first opened the field switch on the machine to be connected and disconnected the main lead of the machine on the side opposite from that which contained the series winding; then after closing the field switch we closed the main switch and let it remain closed for perhaps 15 sec. Next, we opened the field switch again and connected the main lead previously disconnected. On closing the field switch the polarity was found to be correct. This was repeated with the other machines, and we were straightened out by 5 min. past one.

Instructions for correcting the polarity of reversed machines, usually say "lift the brushes so that they are clear from the commutator," but when one is in a hurry to get the machines on the line, the method just described is much quicker.

L. E. THOMPSON.

Medford, Mass.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Utilization of Blast Furnace Gas

By Everard Brown

A blast furnace, when operating normally, will generate approximately 150,000 cu.ft. of gas per ton of pig iron produced. This gas, which is really a by-product, is sufficient for the total heating and power requirements of the furnace itself and still leave considerable that may be used for other purposes.

The tendency in the past few years has been to utilize all, or as much of this fuel as possible, although a few of the older plants still use only enough to heat the blast. The more modern plants utilize all of the waste products possible even to the slag which is now ground up and takes an important part in the manufacture of portland cement. Many furnaces are also favorably situated to furnish fuel for power to adjoining steelworks; very often this is a consideration when building a new plant.

Surplus Available

Consider, for example, a blast furnace producing 500 tons of pig iron per 24 hr. At the rate of 150,000 cu.ft. of gas per ton, it will generate 75,000,000 cu.ft. every 24 hr. Of this amount 10 per cent. must be allowed for losses and leakage; 2.5 per cent. for cleaning, if the gas is to be used in gas engines; and 30 per cent. for the stoves that heat the blast. Summarizing, it will be:

Losses and leakage	7,500,000 cu.ft.
Cleaning	1,875,000 cu.ft.
Hot blast stoves	22,500,000 cu.ft.
Total	31,875,000 cu.ft.

Deducting this from the 75,000,000 cu.ft. generated, leaves 43,125,000 cu.ft. every 24 hr. or 1,796,875 cu.ft. per hr., available for use either under boilers or in gas engines. The thermal value of this gas will range from 90 to 110 B.t.u. per cu.ft., depending upon the character and quantities of raw material charged into the furnace; therefore, assuming an average value of 100, there are 179,687,500 heat units every hour that can be utilized. This is equivalent to 12,835 lb. or 5.73 gross tons of coal at 14,000 B.t.u. per lb. With coal costing, say, $2 per ton, the saving would be $11.46 per hr. or $101,265.60 per year of 365 working days of 24 hr. each. Blast-furnace operation is continuous and is never interrupted except for relining or other extreme causes. If the fuel is to be used entirely for generating steam, the saving would practically be net but, where gas engines are employed, there would be a small deduction, owing to the necessity for additional gas-cleaning apparatus. An item of this size, however, is too large to be easily lost sight of, particularly in view of the present-day tendency toward conservation.

Disposition of Gas

The use of this energy to the best advantage seems to be a problem not yet fully solved; in some instances it is utilized entirely to make steam and in some it is all converted into electric energy through gas engines, and in still others a combination of boilers and gas engines is employed. All three have advantages and disadvantages. Perhaps the most important arguments in favor of gas-fired boilers with steam engines are reliability of operation, lower initial cost, less thorough cleaning of the gas required and less skillful attendance. On the other hand, some of the advantages of the gas engine are greater thermal efficiency, elimination of boiler plant, less ground area required, etc.

Many plant managers are still doubtful about the gas engine, particularly when used as a blowing engine, in spite of the fact that its efficiency is greater than that of the ordinary reciprocating steam engine. They seem to think that the costs for repairs, maintenance, etc., might become excessive, and object to the high first cost of such an installation. It is undoubtedly true that the charges for upkeep and capitalization on a gas-engine installation are greater than for a steam plant, but large gas engines are being developed rapidly and before long will, no doubt, be more favorably regarded than at present. Whether gas engines will be able to compete successfully with reciprocating engine and turbine combinations is yet to be settled, particularly as far as blowing engines are concerned.

Use under Boilers

To develop one boiler horsepower with blast-furnace gas having a thermal value of 100 B.t.u. per cu.ft. will require approximately 525 cu.ft. This is equivalent to a consumption of 3.75 lb. of coal per boiler horsepower, the coal having a heat value of 14,000 B.t.u. Dividing 1,796,875 cu.ft., the total amount available as previously mentioned, by 525 gives 3423 boiler horsepower. Good compound-condensing engines should develop two engine horsepower on one boiler horsepower, therefore, the total power available would be 6846 engine horsepower. Furthermore, to supply the necessary blast for a 500-ton furnace should not require more than 2500 hp., so 4346 hp. would still remain—ample to take care of the skip hoist, pumps and miscellaneous items and also leave considerable for purposes other than the actual requirements of the furnace.

The saving resulting from this use of the gas is equal to the cost of the coal displaced plus the cost for labor and maintenance, as well as interest on the difference in the initial cost of coal-fired over gas-fired boilers, where stokers are employed. The saving in labor is in itself quite an item as there are practically no coal or ashes to handle and no fires to clean. As this saving in the boiler house is due directly to the use of gas as fuel, the blast furnace should be given credit for a certain part of it; that is, credit should be given for the amount of coal displaced less the cost of any additional cleaning the gas might be given over and above that given the gas used in the hot blast stoves. Such extra cleaning would be chargeable directly to the boilers.

Use in Gas Engines

While some appliances and operations in connection with a blast furnace under all conditions require steam, their aggregate is hardly more than 200 boiler horsepower for a 500-ton furnace; therefore, being very small in comparison to the total amount generated, it can be ignored. This is assuming that gas engines are used for creating the blast as well as generating electric power for pumps, hoist, etc.

A gas engine operating normally and with a good load factor will develop 1 hp. on about 150 cu.ft. of gas of 100 B.t.u. heat value. The total energy that can be generated by this means will be,

$$\frac{1,796,875}{150} = 11,979\ hp.$$

Deducting from this the 2500 hp. required for blowing, leaves 9479 hp.—an amount far in excess of the requirements of the furnace. In this case, however, as compared with gas-fired boilers there would be a somewhat larger deduction on account of the extra cleaning and purifying required by the gas preparatory to its introduction into the engines. As for the accounting, the blast furnace should be given credit for the actual number of heat

units delivered, basing the calculation upon their coal equivalent minus the cost of cleaning.

From these figures it can be seen that blast-furnace gas plays a very important part in the economical operation of the furnace and can be made the means of increasing the profits and conserving the coal pile for future and other uses.

Cooling Water System for Suction Producers

By W. E. Wort

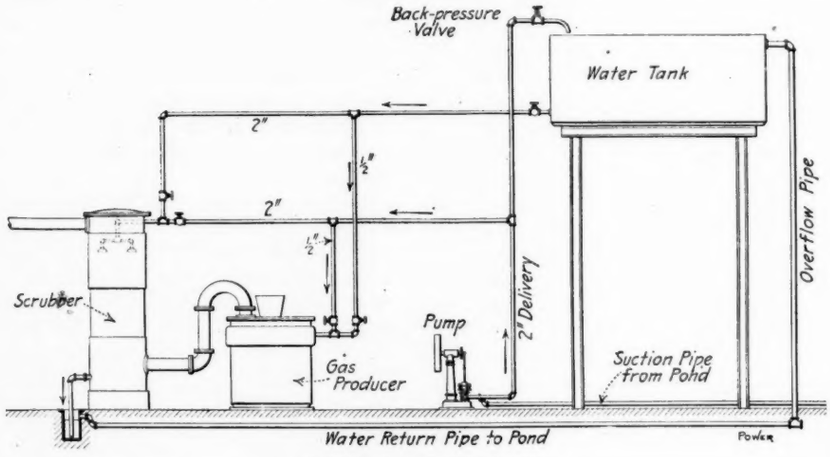
Frequently a drawback to installing a producer-gas plant for moderate power is the difficulty in obtaining a supply of cooling water, or if the water is taken from a town supply the cost is often a large factor. It sometimes happens that a suction producer plant is desired for light and power around a country estate far removed from a town supply of water, and at the same time where only a limited supply is obtainable from the well on the premises. The writer recalls a case of this kind where the capacity to be installed was about 130 b.hp. with an average daily run of 10 hr. The water required by the scrubber and the vaporizer amounted to between 350 and 400 gal. per hour which meant 3500 to 4000 gal. per day. With a limited water supply this was quite an item; hence the problem was solved in the following manner:

While installing the plant, a 50x25x3-ft. reservoir was dug, and into this all the surface water from the land was drained. In order to make the pond water-tight it was first puddled with clay over which was poured a thin layer of concrete and this, in turn, was faced with cement. At a convenient place in the pond a suction pipe was placed in a cylindrical tank which had a number of holes driled around its sides, so as to alow the water to flow in. The suction pipe was led to a pump driven from one of the engines, and this pumped water into a tank placed about 35 ft. above the ground. From this tank a 2-in. pipe was run to the producer house, a ½-in. branch going to the producer boiler and the 2-in. pipe continuing to the scrubber. The supply in this instance was by gravity. It was found that when working the plant to its full capacity that the head of water was insufficient, so a spring back-pressure valve was fitted at the discharge end of the delivery pipe, the spring being set to give a pressure of 60 lb. per sq.in. in the delivery pipe, against which head the pump was always working. A branch pipe was fitted in the delivery pipe on the pump side of the back-pressure valve and led into the producer house, and coupled up to the scrubber and vaporizer in a similar way to the gravity feed pipe. In this way it was possible to use either the pressure or the gravity supply. The pressure supply has been found more efficient for washing and cooling the gas in the scrubber, as the gas is caused to pass through a much finer spray; consequently there is no danger of the gas short-circuiting past the larger jets such as when the water supply is by gravity.

One advantage of having the gravity supply is that it can be used when the pump is not working. In such an arrangement care must be taken that too much water is not used in the scrubber; that is, more than can be carried away by the outlet pipe, or the water will rise too high, and the gas will be delivered to the engines in a wet state, especially if precautions are not taken to dry the gas after leaving the scrubber. Moisture when carried over by the gas is apt to cause misfiring, and will also make the valve spindles gum and stick.

The water after passing through the scrubber is drained into the pond and again passes through the same cycle. One would naturally expect the water to become foul in consequence of being so often in the scrubber, but this does not occur, because the large area of the pond allows the escape of all noxious gases, and the water is frequently reinvigorated by the addition of fresh rain water which is drained into the pond. At no time does the water become more than slightly warm. To prevent any deposit of lime from adhering to the vaporizer a small quantity of boiler compound is added to the supply at this point. Where a producer is not fitted with manholes the top of the producer must be taken off periodically and the vaporizer thoroughly cleaned. It is necessary to clean out the pond periodically, but not more frequently than once in about six years, so that the expense on that score is not very large.

The cooling of the engine cylinders is effected through the ordinary thermo-siphon system, the jacket water passing through a series of cooling tanks.



Cooling Water System

CORRESPONDENCE

Avoiding a Shutdown

Our plant contains two 150-hp. gas engines operating on suction producers and equipped with a hot-tube ignition system, the tubes being heated by a bunsen burner supplied from the city gas mains.

About 9:30 one Monday morning we were notified that the gas supply would be cut off that afternoon and as we had no other means of ignition, a shutdown seemed inevitable. After some pondering, however, I procured two old bunsen burners, fitted with needle valves and stop-cocks, and plugged up the air holes at the bottom. I then soldered them into the bottoms of two quart cans and inserted a piece of candle wick in the burner and around the base of the ignition tube.

About noon I disconnected the gas supply and attached the cans and bunsen burners. The cans were then filled with wood alcohol, a little of which was also applied to the candle wick and the latter lighted. As soon as the base of the tube became sufficiently hot, the flow of alcohol was regulated by means of the needle valve. This alcohol coming in contact with the hot tube was vaporized and burned, thus keeping the tube hot. The cans were protected from the heat by a sheet of asbestos and all the joints were carefully soldered.

The engines were run in this way for about 5 hr. and it required less than a pint of alcohol for the ignition of each engine.

S. G. Rose.

Brockville, Ont., Canada.

Tests made by Prof. R. C. Carpenter, of Cornell University, on a sun-power steam engine, using steam of about 1 lb. above atmospheric pressure and a vacuum of 28 in., showed that it required about 31.6 lb. of steam per hp.-hour.

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Cost of Making Ice in Small Plants

By R. P. Kehoe

The business of ice manufacture is sometimes disparaged and manufacturers of such machinery are criticized for the apparent failure of a few plants. The cause is usually attributed to inefficient equipment or misleading statements on the part of the builders. Occasionally the purchaser is convinced that the plant is properly constructed, but believes his operators to be at fault.

In almost every instance the lack of success arises from local conditions which the most perfect plant could not overcome. This is especially true in installations of small capacity. Large plants are only installed where there is considerable demand which lasts for several months during each year, but small plants may be installed in localities having little population and requiring ice for only a comparatively short period.

In cities of any size there is a certain demand throughout the year. Hotels, saloons, butchers and the like require ice even in the coldest weather. Inhabitants of cities are also accustomed to use ice constantly. But in suburbs, country towns and villages, little or no ice is used during the cooler months. June, July, August and September are practically the only months which can be counted on to produce a maximum demand. Moreover, some days during these months will be cool and the business will fall off. Under such conditions the entire output of a plant may not exceed 30 to 40 per cent. of the full yearly capacity.

If the percentage of yearly output were precisely the same for two plants, one of large and one of small capacity, the manufacturing cost per ton should be and is less in the large plant; but usually the small plant must bear the burden of a low yearly load factor, so that it is actually necessary to make a more careful investigation of local conditions governing demand and selling price in the building of a small plant than a large one.

The daily manufacturing cost when the plant is operating at full capacity gives positively no idea of possible success; such figures are entirely misleading. Many small plants now in operation show little or no profit at the end of each year and the owner cannot understand it when the daily operating cost per ton is only about half the selling price. The cost of upkeep, depreciation and labor during shutdowns and comparatively low yearly consumption are not seriously considered.

Very often in a small business the cost of delivery and manufacturing are confused, and the fact that a loss in one end eats up part or all of the profit in the other, may never be discovered. The average owner of a small ice plant is not often an expert business man and cannot analyze the combination of manufacturing and delivery of ice with much accuracy.

The accompanying tabulation clearly shows how much the yearly consumption will affect the total cost of production. The figures covering labor and fuel represent an average and can be adjusted to suit any special locality, but the comparison is a true one in any case.

In the table figures are given for total yearly production equivalent to three, four, five and six months' full capacity. It must be understood that no plant will simply operate for these periods during the year and then shut down. Maximum capacity is usually required throughout July and August and possibly part of June and September, but the balance of the time will only demand from 50 per cent. down to a very small percentage of the full output. The total amount, however, will in almost every instance equal the full capacity for from three to six months. A plant operating with a yearly load factor of 25 per cent. or the equivalent of three months' production may be

COST OF ICE PER TON IN DIFFERENT SIZED PLANTS AND AT DIFFERENT LOAD FACTORS

Capacity, tons ice per day of 24 hr	5				10				15				20			
Total period of full operation, months	3	4	5	6	3	4	5	6	3	4	5	6	3	4	5	6
Yearly load factor, per cent	25	33	42	50	25	33	42	50	25	33	42	50	25	33	42	50
Investment																
Mechanical equipment complete	$6000				8500				12,000				15,000			
Building	1500				3500				5,000				6,500			
Total investment (excluding land)	$7500				12,000				17,000				21,500			
Daily Operating Expense																
One day engineer	2.50				3.00				3.00				3.50			
One night engineer	2.00				2.50				2.50				2.50			
One day tankman	....				2.00				2.00				2.00			
One night tankman	....				....				2.00				2.00			
Coal @ $3.50 per ton	3.50				7.00				10.00				13.00			
Ammonia, oil, waste and supplies	0.50				1.50				2.25				3.00			
Net operating expense per day	8.50				16.00				21.75				26.00			
Net operating expense per day per ton ice	$1.70				1.60				1.45				1.30			
Total Cost of Operation per Year																
Daily Operating Expense during period of full operation, dollars	765	1020	1275	1530	1440	1920	2400	2880	1958	2610	3263	3916	2340	3120	3900	4680
Half of labor expense for balance of year	608	540	473	405	1013	900	788	675	1283	1140	998	855	1350	1200	1050	900
Fixed Charges: 5% depreciation on cost of mechanical equipment	300	300	300	300	425	425	425	425	600	600	600	600	750	750	750	750
Fixed Charges: 2% depreciation on cost of building	30	30	30	30	70	70	70	70	100	100	100	100	130	130	130	130
Fixed Charges: 5% on total investment for repairs, taxes, water and incidentals	375	375	375	375	600	600	600	600	850	850	850	850	1075	1075	1075	1075
Total annual expense, dollars	2078	2265	2453	2640	3548	3915	4283	4650	4791	5300	5811	6321	5645	6275	6905	7535
Number of tons of ice produced annually	450	600	750	900	900	1200	1500	1800	1350	1800	2250	2700	1800	2400	3000	3600
Total cost per ton ice per annum	4.62	3.78	3.27	2.93	3.94	3.26	2.86	2.59	3.55	2.94	2.58	2.34	3.14	2.61	2.30	2.09
Minimum average selling price to earn 10 % on investment	6.29	5.03	4.27	3.76	5.27	4.26	3.66	3.26	4.81	3.88	3.33	2.97	4.34	3.50	3.02	2.69

taken as a bad case, although this is entirely possible in certain localities, particularly in the Northern states or where ice is only required for a summer population. The equivalent of six months' production would, on the other hand, represent good conditions for an average plant.

The estimated costs of installation are fairly liberal, but can be depended upon to be very nearly correct. The building for the 5-ton plant is assumed to be of very cheap construction on account of the small size of the plant. The larger plants would need stronger and more elaborate structures, which explains why these figures are not exactly proportional.

In the 5-ton plant the engineers can fire the boiler and also pull the ice. For the 10-ton plant, one tankman is figured on to harvest all the ice during the day, as the operation can be arranged in this manner by providing a distilled water-storage tank to collect the condensate during the night. Four men are necessary for both the 15- and 20-ton plants.

During the cooler months the labor expense can be reduced somewhat and in the tabulation, this item is calculated at 50 per cent. of the full expense, which can be assumed as a fair basis. On the mechanical equipment, 5 per cent. depreciation is conservative, because it does not cover any repairs or upkeep. If this 5 per cent. is actually set aside as a sinking fund and invested to earn a normal rate of interest, it would equal the total value of the equipment in about 15 years. In addition, 5 per cent. on the entire investment is calculated for yearly repairs and incidentals.

The cost of property has not been considered and would depend upon the location. This part of the investment really need not be required to earn anything, as generally it will increase in value during the life of a plant. It can therefore be considered apart.

The last two lines are of particular interest. It may be noted that the total cost per ton of ice per year ranges from $2.09 in the 20-ton plant with a 50 per cent. load factor to $4.62 in the 5-ton plant with a 25 per cent. load factor. Furthermore, the selling price necessary to obtain to earn 10 per cent. on the investment ranges from $2.69 to $6.29 per ton. These figures may be astonishing to those who only figure on the reguar operating expense without regard to the other items, but they are nevertheless perfectly true and represent the only proper basis on which to figure profits.

This article is not intended to discourage investors for there are hundreds of very profitable plants now in operation that indicate the success of the business in general. The gradual decline of the natural ice trade occasioned by the condemning of questionable sources of supply and the growing preference for the purer artificial product, promises a still better outlook in the future. A few failures, however, are often considered to the disadvantage of the entire business and a more universal knowledge of the true circumstances should help the situation.

CORRESPONDENCE

Ammonia Absorption Refrigerating System

In the Sept. 24 issue of POWER, Henry Torrance, Jr., published a short discussion on my article on this subject in the issue of Apr. 23, so that it is desirable to add a few words here to complete the data originally presented.

The article was not written to present actual figures of the performance of each apparatus of the system, but to explain in detail their functions, and the figures served to describe them more clearly. As stated in the article, the amount of heat eliminated in the rectifier was determined under the assumption that the partial pressure of the water vapor in the gases leaving the analyzer corresponded to the water-vapor pressure at the temperature of the gases. Lacking more exact information at the time, this method of figuring was deemed sufficiently accurate for the purposes of the article. In view of Mr. Torrance's statement additional data may be of interest.

The partial pressure of the water and ammonia vapors in the mixture of gases evolved from aqueous ammonia solutions were investigated by Perman, who used for his experiments solutions of varying strength ranging from 4 per cent. to a maximum of 24 per cent. of ammonia dissolved in the water. The writer substituted the values found by Perman in the following formula to determine the molecular rise in the boiling point of aqueous ammonia solutions for the concentration used in the calculations.

$$E = \frac{\Delta t M}{m}$$

Where E is the molecular rise in the boiling point of the solution, Δt the observed rise of the boiling point, M the molecular weight of the dissolved substance and m the weight of the substance dissolved in 100 grams of the solvent. It was found, as was to be expected, that E increased with the concentration from 1.5 for a 3 per cent. solution to 4.5 for a 24 per cent. solution, that is to say, the stronger the solution the smaller will be the proportion of the solvent vaporized on boiling. For the calculations used in the text to illustrate my article, the strength of the liquor varied between 18 and 35 per cent. Taking the average concentrations of the strong and weak liquors as a basis, the average rise in the boiling points of the various solutions can be found.

The hot vapors ascend to the analyzer in which they are cooled by the rich liquor coming from the exchanger, and a change in the proportion of ammonia in the vapor takes place. From the temperatures of the vapor leaving the analyzer and leaving the rectifier, the quantity of water vapor that is condensed and of the ammonia vapor that is reabsorbed, can be determined. These figures used in connection with the latent heat of steam and heat of absorption of ammonia for the temperature and pressure conditions existing will give the quantity of heat rejected in the rectifier.

Recalculating the heat rejected in the rectifier by using the above quoted formula as a basis of final figures, of the steam consumption in the generator per hour per ton of useful refrigerating effect, the following results were obtained:

Test No. 1	Test No. 2	Test No. 3	Test No. 4
41	52	35	26.4

These results, although lower than those given in the article, still show the same change in the steam consumption for varying conditions of operation, the steam consumption increasing when the temperature in the refrigerator decreases.

The water consumption was obtained from the actual tests of the particular plant described, and while it may be higher than that of some other plants differently arranged, due consideration was given to this fact in the calculations. The use of large quantities of water for cooling purposes is an advantage when it can be secured cheaply.

It is not proper to compare either the steam consumption or the water used by an absorption refrigerating machine with that of a compression refrigerating machine operated by a simple Corliss steam engine, because the quantities of heat carried off by the water are different in these apparatus. If the actual amounts of steam and cooling water necessary for the proper operation under the same conditions of an absorption refrigerating system and a compression refrigerating system operated by a compound condensing Corliss engine are compared, the results obtained will be of some value.

In view of the above, the final conclusions drawn from the quantative analysis of the absorption system, as given in my article, still stand.

FRED OPHÜLS.

New York City.

Another Diesel engine motor-propelled ship is to be built for service on the Great Lakes for the Montreal Transportation Co. *The Engineer*, London, says the vessel is to have two sets of Diesel engines, each of 300 hp., running at 400 r.p.m. Each engine will drive an alternating-current generator and the current will be delivered to an induction motor on the propeller shaft running at 80 r.p.m. The vessel will have a capacity of 2400 tons on a draft of 14 ft.

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Sworn to and subscribed before me this 30th day of September, 1912.

Wm. W. Mayer,
[Seal.] Notary Public
(My commission expires Mar. 30, 1913)

For the contents of this issue, refer to the Outline of Topics, page 62.

POWER

Defective Boilers

Regarding boiler explosions, their cause and prevention, the equivalent of several large volumes has been published in POWER in recent years. Engineers and firemen have been advised as to their care and operation, but in spite of all that has been written and notwithstanding that boilers are better built and cared for than formerly, between thirteen and fourteen hundred serious boiler accidents occur in the United States every year. Of these, according to William H. Boehm, in a recent lecture, abstracted on page 548, between three and four hundred are violent explosions, in which between four and five hundred persons are killed and from seven to eight hundred more are injured, and more than half a million dollars worth of property is destroyed.

During the past ten years 1,101,140 boiler examinations have been made by one boiler-insurance company, out of which 140,989 were reported defective, many of which were considered dangerous, and one boiler out of every eight examined contained defects of such nature as to be reported by the inspector.

These figures are of such magnitude that those responsible for the upkeep of steam boilers should exert greater vigilance in their examination and know, insofar as it is possible, that defects do not exist.

If one boiler out of every eight examined is defective in one way or another, then the ratio is greater, when the total number of boilers in daily operation is considered. Thousands of them are never examined by other than the men in charge, who, in many cases, are not qualified to intelligently determine whether a boiler is defective or not.

It goes without saying that many such boilers are operated with more or less serious defects existing which would be detected and ordered repaired if proper inspection were made.

If a boiler is used in a manufacturing establishment, the workmen are exposed to a hidden danger of which they are ignorant and against which they cannot guard. The engineer, therefore, has a double responsibility, because he not only is expected to safeguard the boiler, but the workmen around it.

If he is incompetent the responsibility should, but seldom does, fall upon the owner. If, for the sake of saving a few dollars a year in wages, he hires a man incompetent to intelligently care for the boiler, he should be held criminally negligent if an explosion does occur.

Boilers are dangerous enough at best, the men operating them are often inclined to be careless and allow defects to remain, and the owners, judging from the boilers found in operation, are indifferent as to their condition.

It is about time for everybody concerned to wake up to the gravity of the situation.

The Engineer's Duties

Speaking about putting worn apparatus in working condition, an engineer made the remark that he did not "fool" with anything of the kind because the company he worked for did not appreciate it. They only paid him for operating and not for the time spent in "fixing things."

Is that the right attitude for an engineer to assume, even if the management does not seem to notice those little extra things he does about his engine and boilers, etc.? By doing willingly such little things he shows an interest in the upkeep of this end of the plant that is much to his credit. It does not speak well for an engineer if he allows "junk" to accumulate, and when an injector, valve or anything of like nature is disconnected, because some part is worn or broken, he should put it in shape, or have it done as quickly as circumstances will admit, when it can be laid away for future emergencies.

Such a rule, if adopted by all engineers, would materially assist them in their work, at times, as well as save time and money for their employers.

This is not to say that any engineer should take the responsibility, or more properly speaking, the burden upon his

shoulders of doing all the repairing, whatever the size or nature, required in the plant, but that he should give attention to little things he can accomplish unhampered while on duty, and leave such work as is justifiable to a machinist.

An engineer should not do heavy repairing, simply for the reason that he is not paid to do it, taking into consideration the general wage received by engineers, his duty ends with operating, and seeing that things are always kept in the best operating condition, but this may require that he do many little jobs without the aid of an outsider. In so doing he develops his own resourcefulness and shows his ability to advantage.

The foregoing applies only to the average plant. In the larger ones things are carried out more systematically.

While there is a limit to the amount and nature of repairs that should be done by an engineer, he should not be too ready in "drawing the line," as an old chief puts it. It requires judgment to know where to begin and stop such work.

Scarcity of Elevator Literature

Nearly all, if not all, the various branches of power-plant engineering are well represented in engineering literature except that pertaining to elevators. By this statement we do not mean to infer that the few books devoted to this subject are unworthy of commendation. We do deplore the lack of such literature.

In modern city life the elevator is an extremely important factor. It is said that more people travel on elevators in New York City in one year than in all the subway, surface and elevated railroads combined. The problem of safely carrying so many passengers is serious.

In large office buildings the care of the elevators is only a part of the engineer's duties. True, the watch engineers in the very large office building plants have little to do directly with this work as it is entrusted to the care of a foreman and crew who do nothing else. But, the chief engineer is responsible to the management and the public for his foreman, and watch engineers are future chiefs.

If it were not for the work of trained inspectors of the insurance companies the number of fatalities due to wrecked elevators would be startling. This is no reflection upon makers of elevators or the men caring for them. The latter do well —the wonder is that they are so successful. In seeking to learn more about their work these men look for books covering the subject, and are astonished to find that there is no complete work available. The elevator makers are none too generous or responsive to requests for information concerning their machines. There is one commendably complete book on hydraulic elevators, but, to the best of our knowledge, none thoroughly treating electric machines. Some engineering books make a pretense of treating them, but the information given is far too meager to be of much value.

We were fortunate to obtain and publish during the years of 1908 and 1911 inclusive, a series of articles on electric elevators by the late William Baxter, Jr. This series made up the most thorough and practical treatise on the subject ever published. But this is not enough. A treatise on elevators embracing all kinds is sorely needed.

A Plan Worth Adopting

With the approaching season many engineering organizations will begin some sort of an educational program for the year. Here is a plan that all such bodies might advantageously adopt with confidence that much benefit will be realized.

Number Three Association (Newark, N. J.) of the National Association of Stationary Engineers has delegated two of its members to make plans and specifications for modern power plants. One is to cover the requirements of a high-class office building which, among other essentials of equipment, will have a refrigeration plant, coal and ash conveyors, vacuum-cleaning apparatus and the highest type of air purification and ventilating system. The other will design a first-class factory power plant. Each man may ask the collaboration of any others, but the aim is to make the work solely that of this local association.

Designing power plants is splendid training for operating engineers as it brings them to understand the logic of many things they had come to regard as matter of fact from long familiarity. There is every reason why a first-class operating engineer should be able to design a plant that would prove thoroughly practical and economical in service. His experience in operating power-plant equipment is very helpful to him as a designer. Making the specifications would give him a training in detail that should be of inestimable value in his regular work.

It is to be hoped that other associations having to do with power-plant practice will take some suggestions from this plan and incorporate them in their winter's educational program. That the benefits from it may be participated in by all the members, it would seem a good suggestion to have lantern slides made showing the work as it progresses and keep notes of the changes in the plans and the reasons for making them. Those in charge of the work could then give illustrated talks before all the members and thus insure the fullest realization of the purpose. It would make the plan still more fruitful of good results if a number of similar associations would engage in a friendly competition to see which could produce the best plans and specifications. Judges could be appointed to pass on the entries in the competition and award prizes for the best work similarly as in intercollegiate debate or athletic meets.

Smoke Abatement

The monthly meeting of the New York section, American Society of Mechanical Engineers, on Oct. 8, is to be devoted to a discussion of smoke abatement, and representatives from other cities will report on what measures are being taken to handle this problem in their localities.

While the smoke nuisance in New York has not become as grave as in some other cities because of the general use of anthracite, still considerable bituminous coal is burned and without regard to smokeless combustion. In proof of this, one has only to survey the city from the top of a high office building on a humid day. As already mentioned in these columns, New York City has no separate smoke-inspection department, this duty being performed by the Health Department, but it seems to have difficulty in obtaining the conviction of offenders.

Heretofore, there seems to have been a tendency to side-step the issue on the grounds that the situation was not acute enough to warrant agitation. The fact remains, however, that with proper handling, bituminous coal can be burned without obnoxious smoke, and it is the duty of the community to see that such measures are enforced. In taking the initiative the local committee of the American Society is to be commended.

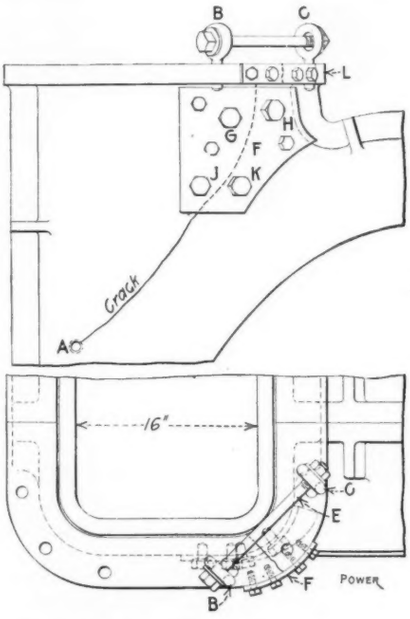
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Patching a Pump Case

Owing to the failure of a wharf crane, a new pump case was dropped several feet and cracked badly at the throat, as shown. It was a bad spot for a crack, as the case joints were overhung; that is, the joint was inside the bolt circle and screwing up on the bolts tended to open the crack still further. To be of any use the patch had to be air-tight. The repair was made as follows:

The full extent of the crack was determined by first rubbing the surface of the casting with waste soaked in kerosene and then wiping it clean and rubbing



How the Pump Case Was Patched

chalk over the face. The crack appeared plainly under the chalk. The lower end was drilled for a 5/8-in. screwed plug *A*.

As the top of the crack had sprung open 3/4 in. and the case was 1 3/8 in. thick it was a large crack. A couple of eye-bolts *B* and *C* were forged and the shanks turned 1 1/4 in. diameter to just drive lightly into the holes in the case flange. A 1-in. bolt *E* was passed through the eyes and washers and tightened until the crack closed.

After being bent well to shape and served with red lead paint, a plate *F* 1/2 in. thick was put on with capscrews, as shown. The holes *G*, *H*, *J* and *K* on each side of the crack were countersunk on the outside and the screws used on them were taper-headed at the bottom of the hexagon to fit. A slight draw was allowed on these case holes so that the screws drew the crack shut when pulled tight. A supplementary capscrew patch *L* was put on the edge of the flange also. This completed the job. A larger patch would have been better, but the above was perfectly satisfactory under working conditions.

J. McA. Howden.

Melbourne, Australia.

Boiler Water Impurities

Much has been said about the way we should obtain a perfect analysis of our boiler scale, feed water and the manner of feeding compounds and the proper kind of compound for each individual case to be treated. It seems to me that the first thing to be considered is the proper removal of the scale from the boiler. The treatment of the water before it enters the system is important, but the purifying effects of the feed-water heater and the mingling of the returns with the feed water before entering the boiler should not be forgotten. The scale should first be removed before introducing a compound.

Let us trace the water taken from a running stream. It is pumped to a reservoir and the mud is settled, nearly all being left in the reservoir. It leaves the reservoir and enters the open heater and, by being heated to 200 or 210 deg., most of the foreign matter is caused to adhere to the interior of the heater. As the heating takes place, another change of great importance, often overlooked, is the addition of perfectly pure water in the form of condensation, which in some cases forms a large part of the entire feed water leaving the heater. Then the steam-trap returns are also very important. If there is a purifier (scum-cock) on the boiler it will remove about one-sixth to one-eighth of the impurities if properly used, after the heater has removed three-fourths. Thus, the water enters the boiler very nearly pure.

Now, if the boiler has a general deposit of old thick scale the thing to do is to remove a large thick piece and have it analyzed. It might be well also to send a gallon of the water from the suction of the feed pump. I never feed compounds into the water before it enters the heater, as I believe it interferes with the purifying effect of the heater.

Ray Gilbert.

Virginville, Va.

Care of Electric Elevator Brakes

This is a summary of my experience with the troubles and dangers to the brake of a common type of electric elevator.

When the starting lever in the car is moved to the up or down motion, it sends current through the elevator switchboard, through the circuit-breaker and the soloneid *B*, which draws up *C* to *D*, compressing the spring and slackening the brake bands as shown dotted. When the controller is in the "off" position all current is cut off from the solenoid. Then the tension of the spring causes the bands to grip the brake wheel as shown by the full lines.

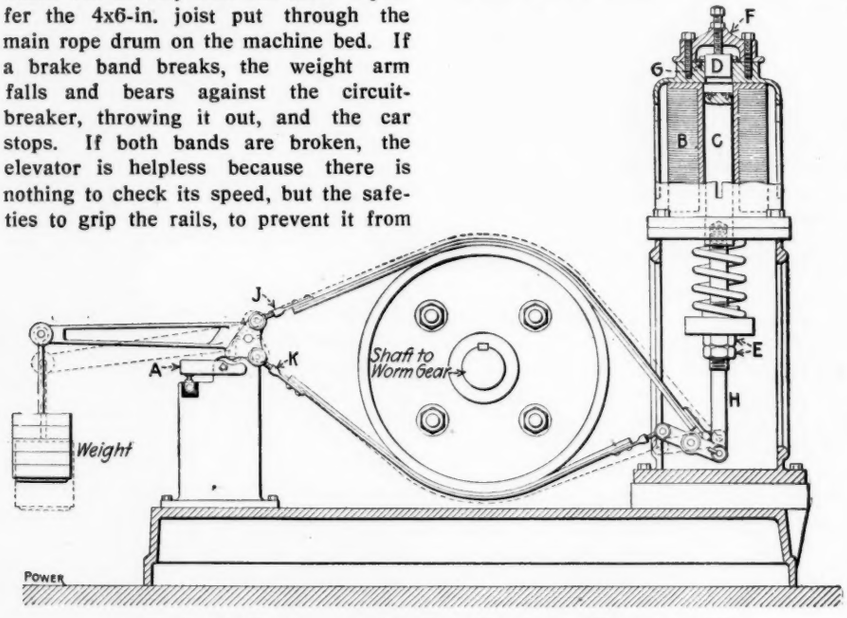
My greatest trouble with these elevators, which run 415 ft. per min., and in fact all high-speed electric elevators, is to make a good stop. The stop should not be too abrupt and yet the car should not slide, as both are equally dangerous. If the stop is too sudden it is likely to set the safeties, and if too much sliding occurs the car will go by the floors, causing a restarting which uses up more current than when running three stories. If the car slides excessively it is likely to go to the roof and against the limit switches, cutting off all current by throwing out the circuit-breaker *A*, and if the counterweights are slightly long, they will pound hard on the floor. When the stop is too abrupt, a piece of yellow soap held against the brake wheel is a good remedy; it will keep the car from sliding. Powdered rosin is good in extreme cases, but good judgment must be used with both. Elevator inspectors do not approve of using anything on the brake wheel.

With speeds of 250 ft. per min. rosin or soap is unnecessary, but with a speed of 415 ft. the brake band will get so hot it will smoke when the trips are frequent and the brake will slide as though greased.

When the foregoing remedies do not correct the trouble it is necessary to change the brake connections by screwing up on the nuts *E* under the spring, giving more brake friction, which in time, due to the leathers wearing, settles down and the plug *C* drops away from *D*, leaving too much space between them. The result is that the magnetizing effect is decreased and the brake bands will drag. The remedy is to take off the top *F*, raise the plug *D* and put thinner

liners in the place of those shown at *G*. This brings the plug *D* nearer plug *C* and increases the drawing power to lift *C*. Care should be taken not to get *D* and *C* too near together, as there will be no chance for *C* to lift because of coming in contact with *D* before the brake bands have lifted away from the brake wheel. The main switch should be pulled before attempting the foregoing.

Another bad condition is when the rod *H* strikes on the bottom bed, thus holding the brake bands loose. This may be caused by thin leathers, or the adjusting ends *J* and *K* being let out too far. The remedy is to take up *J* and *K* equally and renew the leathers. In most cases the rod *H* will have to be screwed into *C* to maintain the original space between *C* and *D*. Before doing this be sure to block the main rope drum by putting a 4x6-in. joist through the spokes. Then by letting up on the brake the car will not go flying to the roof, as it would do because of the action of the back drum counterweight. Another way is to run the car to the roof and pull in the safety, thus holding it. Still another, when the car is at the top, is to put a 6x6-in. block under the bottom counterweight, which will be very near the floor. I prefer the 4x6-in. joist put through the main rope drum on the machine bed. If a brake band breaks, the weight arm falls and bears against the circuit-breaker, throwing it out, and the car stops. If both bands are broken, the elevator is helpless because there is nothing to check its speed, but the safeties to grip the rails, to prevent it from racing to the roof with a light load, or racing to the bottom if heavily loaded.



SHOWING CONSTRUCTION OF TYPICAL ELECTRIC ELEVATOR BRAKE

After changing the band connections at *J* and *K* there is probably either a sudden stop on the up motion and a sliding stop on the down motion. One would think that by tightening one connection the friction would be equal, but such is not the case. To loosen the connection to a solid stop on the up motion, the top brake band must be loosened by taking out the bolt and giving it a half turn and if sliding on the up motion occurs the top band should be tightened; if the down stop is too abrupt loosen the bottom band.

To put on brake leathers select a piece of fine grain leather about ¼ in. thick and of even thickness. Rivet one end to the band first, then press it down hard on the iron band to get the holes marked on the opposite end. The other holes can be bored by putting the drill through the holes in the iron brake band. The leather holes should be countersunk ⅛ in. for rivet heads.

Revere, Mass. A. C. WALDRON.

Boiler Explosions

Permit me to commend the article under the above heading by S. F. Jeter in the Mar. 26 issue, and the editorial comment on the same in the Apr. 9 number which was also interesting. All should particularly read the two articles above mentioned, together with "Uniform Boiler Specifications" in the Apr. 2 issue.

I like a suggestion I found in the latter article, which read "Institute an educational campaign" relative to boiler construction and attachment of appurtenances, etc.

With the high pressures now employed I believe a more radical departure from old times should be made than is being done just now. What is needed most is a gradual reduction in diameters, and all flat surfaces eliminated if possible. In firebox boilers it is usually the crown-sheet which gives way, while in water-tube types it is the drum-heads.

By unequal expansion when under steam, it may be all this part of the boiler can do to withstand the pressure until finally a slight jar from water-hammer, etc., may cause it to let go when it is least expected.

I think the Jacobs-Schupert firebox an ideal departure from the old stay-bolt method of firebox boiler construction judging by the report of tests made on it, since it makes this part of the boiler practically as strong as the shell at all times.

To keep track of the age and firm turning out boilers, it would be well to require that an indestructible plate be riveted on boilers giving the year made, whom made by, strength of material, etc., or if objection should be made to this plate, the same might be stamped upon the boiler in some conspicuous place where it would not become obliterated, as prescribed by some state laws.

While I am upon the subject I wish to ask, if it would be of any benefit to "draw" boiler shells as seamless flues are drawn?

LLOYD V. BEETS.

Nashville, Tenn.

[The contributor's question relative to the "drawing" of boiler shells is indeed interesting. We would be pleased to publish good, live comment on it in which should be stated the mechanical difficulties to be met with in doing this work and the probabilities of it as a commercial success.—EDITOR.]

Valve Guide in Pipe Damages Engine

A serious accident happened to the high-pressure cylinder of a Corliss engine I have under my supervision. The piston was broken into about 150 pieces and both cylinder heads seriously fractured. The damage was caused by a valve guide of a 4-in. globe valve, which is an auxiliary valve placed in the expansion loop above the engine. The valve guide had been broken off sometime previous to my taking charge of the plant and had lodged in the steam chest; otherwise it would have been noticed as I have had the line from the valve to the engine all apart repairing leaks.

The crank end head broke first and I was trying to throttle the engine, but before I could do so I had to jump to save myself from the flying fragments and escaping steam, and shut the stop valves on the boilers to save the siphoning of the water from the boilers to the air compressors and other engines then in operation. I hope that any engineer missing part of a valve or other fitting in a steam line will take heed and, before damage results, find where the part has located itself. Should this case have had that attention, I would not have had this experience.

I had considerable trouble with a dashpot on the high-pressure cylinder during my first week in charge of the plant. The plunger did not fit snug enough. I found by putting oil around the plunger it would work and with a very strong vac-

uum. I then tried graphite and cylinder oil, putting it on with a brush before starting each day, and this ended my dashpot troubles.

JOHN T. CHAMBERLAIN.

Irvington, N. J.

Jointing Materials

Conditions of use determine the choice of a jointing material for any particular purpose. The condition of flanged faces to be brought together to stand various pressures and the nature of the liquid or gas under pressure, together with temperature, are the chief factors determining what to use.

In early steam-plant practice when 5 to 10 lb. pressure was a maximum and when the engineer's chief difficulty was to prevent a vacuum in the boiler, oakum was the material largely used. A leak in a pipe line has many times been stopped with a piece of canvass and tarred yarn when the pressure did not exceed 10 lb.

Later, with faces perfectly scraped to surface plates, a wipe of tallow or boiled oil to prevent rust was sufficient to rigidly couple two cylinders together without leakage. Now the tendency is to consider the jointing material and finish surfaces accordingly.

Where cold water under pressure only has to be considered, ordinary hard woven paper millboard of suitable thickness makes excellent joints and is largely used. The surfaces in this case are left as machine finished with a coarse feed. Fire-engine makers use a more expensive material—leather—for covering joints, this, needless to say, makes first-rate joints, but is expensive.

My opinion, borne out by considerable experience, is that clearance volume in steam cylinders is frequently excessive with consequent loss of economy, due to bad jointing. I have seen cylinder covers in a big job with joints ¼ in. thick under them. A distinct improvement in economy can be made by looking to this point and substituting a thinner material. The designer as a rule does not take this fact into consideration when figuring the probable results of a new engine over the drawing board. With a triple-expansion engine of 2000 hp. a good many extra tons of coal can be wasted annually by thick cylinder-cover joints. Broad face cylinder-cover joints, where the pressure does not exceed 80 or 100 lb. with the surfaces in good condition can be efficiently made with thick brown paper steeped in boiled oil. Above this pressure the thinner asbestos millboard which will make a tight joint should be used, or one of the numerous patented jointings of the thinnest character. Much depends upon the closeness of pitch of the studs. It is better that these be smaller in diameter and closer together than of large size coarsely pitched.

With higher pressures like those used in freezing machines, it has been found advisable to use metal-to-metal joints, with flanges having grooves to take a solid unbrazed ring of annealed copper, which forms an ideal joint.

Corrugated gaskets and red lead make good joints, but care must be taken that the gasket is the full size of the flanges. If simply put inside the bolt circle the flanges suffer distortion and are a nuisance every time they are uncoupled and again have to be jointed.

Thick joints are only necessary where the surfaces are in bad condition. The worth of a jointing material depends upon its elasticity under the conditions of its temperature and use. As a rule cover joints of any description are not made to be permanent and where studs are used, excessive tightening down stresses them unfairly. Rather than attempt to squeeze too tightly when making cover joints with sheet material to hold steam, it is better to go over the bolts and tighten them after the joint is warmed. The least pressure possible to just stop leakage is the desideratum.

To satisfactorily discuss the various patented jointing materials and their claims would require considerable space and serve no useful end. They are legion and in the main serve the purpose for which they are made; but in some instances an ordinary material, at a fraction of the cost, would serve many purposes for which expensive jointings are used. Common sense is useful in this matter as in every part of engineering and I take it that jointing with brains is as useful as designing with brains; it is possible to save money both ways.

A. L. HAAS.

London, England.

Carelessness Causes Pump Accident

The chief cause of the accident herein described was due to the fact that "somebody forgot."

A small gasoline engine was used to drive a triplex power pump used to pump oil. Repairs were being made to it by the engineer and a helper. This helper, with all good intentions, threw in the clutch to turn the pump over to the most convenient position for disconnecting the connecting-rods from the cranks. After disconnecting the rods at the cranks only, he walked away leaving the clutch in.

Later the engineer, not knowing the rods were disconnected or that the clutch was in, flooded the vaporizer, threw in the switch and cranked the engine. At about the time he realized the clutch was in, the engine "kicked off." As the cranks revolved they struck the disconnected rods, causing serious damage to the pump.

It pays and pays well to look a machine over carefully before starting it.

J. P. COLTON.

Ohio City, Ohio.

Books for the Engineer

Just what books engineers should have is open to a lot of controversy.

Many older engineers do not believe in the book-engineer or his books, but this is not the case with the men that are coming into the business now. Here in Massachusetts, where the license law forces the engineer to study, a good book is always appreciated. I have personally invested more than $300 in technical books in the last 10 years, a great many of which have been useless to me. I have never, however, found any difficulty in disposing of these latter books, and the fact that a book, or for that matter anything, is of no value to me does not mean that someone else cannot find a use for it. I have found books also that I would not part with under any conditions.

The following comprises a library that from my viewpoint is invaluable to the operating engineer. If I was sure that I could not replace it, a week's pay would not tempt me to part with one of these books.

Each of the following books paid for itself before it had been in my possession six months:

Book	Author
Boilers	Power Hand-Book
Power Catechism	Power
Steam Heating and Ventilation	Technical Pub. Co.
Pipes and Piping	Power Hand-Book
American Engines	T. Hawley
Valve Setting	Collins
Elements of Steam Engineering	Spangler and Green
Steam Engine Indicator	Low
Shafting, Pulleys & Belting	Power Hand-Book
Steam Turbines	French
Hydraulic Elevators	Baxter
Pumps & Hydraulics (2 vol.)	Rogers
Condensers	Low
The Slide Rule	Ramsey and Hinkley
Practical Talks on Electricity (2 vol.)	Baxter.
Motor Troubles	Raymond
American Machinist Hand-Book	Colvin and Stanley
Millwrighting	Hobart
Mechanical Drawing	Willard
Automobile Hand-Book	Homans
American Cyclopedia of Engineering (7 vol.)	Technical Pub. Co.

It will be noted that I have left out such an important subject as refrigeration. I have never found a book that to my mind thoroughly covered the subject. As regards electric elevation, I have Baxter's articles on this subject published in POWER some two years ago, and I have never seen a book that covered the subject as well. As the tendency seems to be toward a more general use of the traction type of electric elevators, Mr. Barter's articles made up in book form would be highly desirable.

As far as compressed air is concerned, the Encyclopedia of Engineering covers the subject well enough to supply my needs.

A. D. PALMER.

Roxbury, Mass.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Counterbalancing Elevators

The following may give the information asked by L. A. Trowbridge in the Sept. 10 issue on the counter-balancing of elevators and the care of electric elevators.

The function of counter-balancing is to minimize the power necessary to operate the car. As counter-balancing depends to a great degree upon the gearing used, this subject will be briefly considered.

The gearing of an elevator is essentially the same as the arrangement of the common rope and pulley block. A one-to-one gear shown in Fig. 1 allows the car *A* to travel the same distance only as that of the piston *B*. Such a gear is really of no practical value on account of the limited travel of the car compared to that of the piston. The height at which the stationary sheaves *C* must be placed is also a serious objection.

Fig. 2 shows the six-to-one gearing of an inverted plunger, high-pressure elevator commonly used in modern high buildings. Here the cylinder *E* is short and the plunger travel very small compared to the rise of the car. For a plunger travel of 5 ft. the car will travel 30 ft., the motion being multiplied by the passage of the cables over the movable sheaves *F, G* and *H* and over the stationary sheaves *J, K* and *L*. Gearing in practice runs from 2 to 1 to 8 to 1, according to the height of the lift. To find the gear of any elevator, add the number of cable lengths on both sides of the movable sheaves. Gears are made even to avoid an undesirable arrangement of the sheaves; the three-to-one gear being the only exception.

Most of the counter-balancing weights are located in the frame which guides the traveling sheaves and assist in raising the car. The empty car must weigh enough to pull up the plunger, sheaves, frame, etc., and also to overcome the friction of all the moving parts. Its weight when empty, is more than enough to do this, however, and, instead of making the car lighter, the weights *M* are loaded on the plunger.

It is here that the gearing determines the weight necessary to counter-balance the car. If it is a two-to-one gear, to balance 1 lb. of car will require a 2-lb. counterweigh at *M;* if a four-to-one gear, 4 lb.; a six-to-one gear, 6 lb., and so on. To minimize the weight necessary at *M*, a sheave *N* over which passes a cable connected to the frame of the car and on which hangs the independent counter-balance weight *O* is provided. As this weight is geared one to one with the car, 1 lb. of weight therefore balances 1 lb. of car. This weight is small compared to the weight loaded on the plunger. The reason for this is that a heavy weight at *O* would have momentum enough at high speeds to teeter the car up and down considerably and also put severe stress on the cables when the car was suddenly stopped. This does not apply to

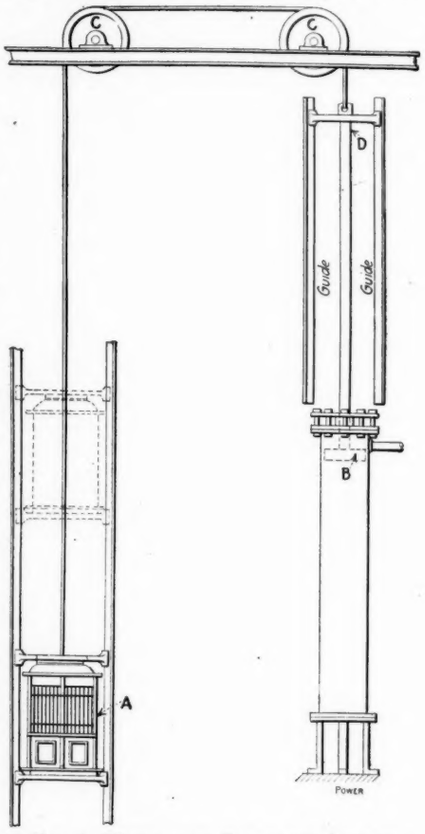


FIG. 1. HYDRAULIC ELEVATOR GEARED ONE TO ONE

the weights when placed at *M*, even though the weight is much more, because the force must act through the gearing before it affects the car. The cable carrying the independent weight *O* also aids in making the machine safer as this cable takes some strain from the other lifting cables. Some installations do not have the independent weight.

Besides the car, the cables in the shaft above the car must be counter-balanced. This may be clearly understood from the following:

Suppose the unbalanced weight of the car when it is at the top is 400 lb., then when it is at the bottom the weight of

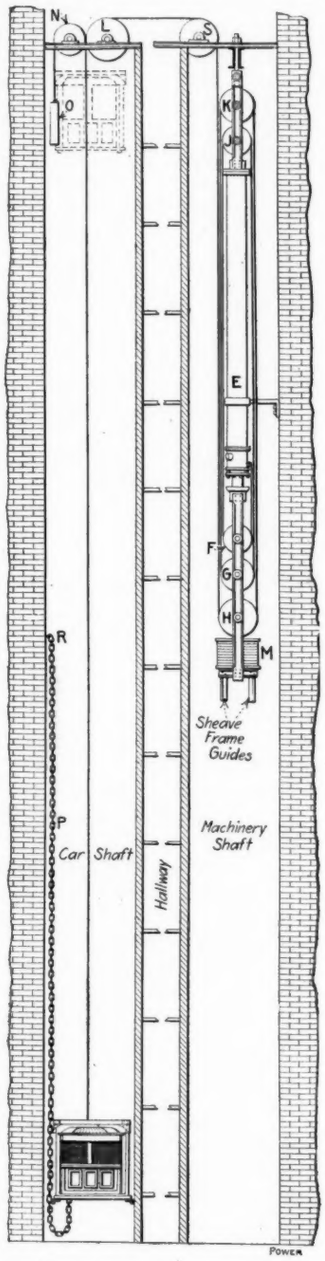


FIG. 2. HYDRAULIC ELEVATOR GEARED SIX TO ONE AND COMPLETELY COUNTER-BALANCED

the cables in the shaft must be added to this 400 lb. These cables are ⅝ in. in diameter and weigh approximately 0.8 lb. per ft. As there are usually four

or six of them, their weight becomes important, especially if the lift is high as it is in most city buildings.

It has often been suggested that the chain *P*, which is used to counter-balance the cables and which is attached to the bottom of the car, be connected at the other end to the independent weight *O*. This would reduce the length of chain, for when the car is at the top the weight would be at the bottom and *vice versa*. This is not wholly practical as there is danger of serious accident, due to the weight getting entangled in the equipment in the shaft. By attaching the chain a trifle above the middle of the car travel, as at *R*, it hangs from *R* when the car is at the top. The chain must weigh as much as the greatest length of cable in the shaft added to that of the cable length on the machinery side of the sheave *S* and from *S* to *H* when the car is at the top of its travel.

The care of electric elevators may be summed up briefly as follows:

Keep all contacts clean and all guides, sheaves, governor, worm-gear, etc., well oiled. Do not allow too much air gap in the magnets. Avoid all unnecessary and uncalled for sudden stops. Open the main switch or circuit-breaker if the car is to lay idle even over night; always do this when making repairs of any kind. Inspect the safeties and all contacts at the magnet control board frequently. Obtain wiring diagrams of the various circuits and learn them thoroughly. One of the best things to remember is not to pull the whole machine apart just because it will not start; inspect the contacts first as it is there trouble usually begins.

L. E. Tavor.

Arlington, N. J.

Mr, Trowbridge should particularly care for the commutator, knife switches and magnets on electric elevators as they require most attention. Where the motor-armature shaft is directly connected to the worm-gear shaft, the armature is reversed in rotation, which is severe on the commutator and brushes, and all troubles common to motors must be met. The magnetic brake on the armature must not be allowed to get out of order, the leather shoes should be renewed if the rivet heads show any sign of striking the wheel as they will cause the brake to slip. The oil in the gag-pot must not be so thick as to retard the action of the brake, nor so thin as to make the brake act suddenly, as this would cause unsteady running of the car. The density must be selected to meet the temperature of the room and in accordance with the conditions under which the machine is operated.

The starting rheostats and automatic or series magnetic controllers must all be kept clean, dry and not allowed to collect dirt or grease around their bases or they will become grounded and heat or burn out. No one special part or parts of these machines requires extra attention, but they must all be gone over regularly and sandpapered or oiled as the case may require. A general inspection of the safety devices on the cars each morning is the only precaution one can take to safeguard the public who must use the elevators.

R. A. Cultra.

Cambridge, Mass.

There are two general types of electric elevators, the worm-gear and the traction type. The traction type is divided into two classes: the full traction and the two-to-one type. In the full traction elevator there are two drums, one directly connected to the motor shaft and one just below it. The cables from the car pass over the motor drum, around the lower drum, up over the motor drum and down to the counterweights. In this type of elevator a small drum is used with a very slow-speed motor, running from 60 to 90 r.p.m. and as the cables only make one wrap around the drum the height of the building does not affect the size of it.

In the two-to-one type the cables are fastened at the top of the shaft and drop down and around a sheave on the top of the car, then up and around the motor drum and down to the counterweights. The speed of the car in this type is only half the speed of the cables. The worm-gear type is too common to need description.

The several ways of counter-balancing include that in which the car is fastened to one end of the cables and the weight to the other as in the traction type. The drum type has two weights, one connected by cables to counter-balance the drum while the cables from the other weight pass over the drum and connect with the car. The car weights should be placed above the drum weights. If they were placed below and the cables holding the drum weights should break the added weight might rush the car to the top of the shaft with serious results.

In high buildings, to overcome the weight of the cables and still keep the car balanced, chains are used which are connected to a point midway in the shaft and the other end to the counterweights, or the chain may be connected from the car to the counterweights. As the position of the car changes the length of chain hanging from it changes and thus makes up for the weight of cables lost by the car ascending. In some cases the chain is connected to the bottom of the car and the other end to the bottom of the shaft. As the car raises, the chain is picked up and having the same weight as the cables, keeps the car balanced at all times. The car is over-balanced to about the weight of the average load. Too much counter-balancing causes the car to jump when it is stopped or started. To prevent the rattling of the chains a light rope is threaded through the links, being cut into lengths of about 12 ft. to facilitate the work.

The controller is usually a magnetic type or one operated by a solenoid which makes connection by a carbon brush passing over copper contacts. The more frequent troubles with the latter type are dashpot troubles, the valve in the bottom of the pot being caught open, allow the arm to raise and cut out the starting resistance so quickly that a fuse is blown, the dogs set if on the down trip, and in any case the passengers are given a severe shaking. If the starting arm is caught up by rough contacts the same thing will happen. The dashpot, valve and contacts should be cleaned, and it should be seen that the arm works freely.

Another trouble occurs when the contacts that close to open the brake, if an electric one, do not make good connection and release the brake. They should be cleaned and the brake should open just after current begins to flow into the armature. If the brake opens too early it may, with a heavy load, allow the car to settle a little before the armature current is sufficient to raise the car. If the brake is set to open too late, it causes the resistance to heat and the car to start with a jerk. If the brake is mechanically operated from the car care must be used in setting it as with new leathers the brake may be late in opening.

In the magnetic type of controller when the first contact is made the circuit is closed through the lifting magnet of the second contact and so on. When a contact fails to close it is customary to look to the contact ahead of it. The circuits in a magnetic-type controller are usually divided and fuses placed in each circuit; these sometimes blow and cause that particular contact to fail. The most common trouble arises from poor contacts and is due to loose connections, burned contacts or dirt getting between the connecting surfaces. The limit switches in the shaft being seldom used may become stiff and refuse to close. The slow-down device on the top of the car may stick open and prevent the car moving. The small cables connecting the car-operating switch and controller may be broken and, as there are usually several dead conductors in the cable, one of them may be used to replace the broken one. The easiest way to find the ends if they are not marked is with a bell and battery.

Nearly all the motor troubles are caused by dirt getting on the commutator or around the brush-holders and care should be taken to keep them clean. Connections sometimes work loose and cause much trouble. The entire equipment should be gone over at least once a day as nothing will injure a building's reputation, especially if it is an office building, so quickly as poor elevator service. Care

should be taken that all grease cups on the sheaves are working and that there is sufficient oil in the worm case. (Castor oil is recommended.) An occasional coating of raw linseed oil and plumbago will lubricate the cables and keep them from rusting.

THOMAS H. WATSON.

Chicago, Ill.

[Mr. Waldron's letter on page 540 of this issue should be read in connection with the above letters.—EDITOR.]

Steam Ejectors on Shipboard

It is not very creditable, as intimated in POWER'S editorial "The Inconsistency of the 'Titanic' Disaster," to Twentieth century engineering to have a large liner slowly sinking because the water could not be pumped out fast enough, while, at the same time, her huge engines were shut down, the fires banked, and boilers blowing off because there was no provision to utilize this enormous amount of energy for pumping purposes.

Separate reciprocating pumps would, on account of their huge size, be out of the question, and centrifugal pumps mounted on the main engine shaft would possibly be open to the same objection. There still remains, however, the lightest and simplest of all known pumping schemes—the steam jet. It takes up scarcely any room, has nothing to get out of order, can be instantly put in action and is self-priming. Any number of these ejectors can be connected to the same steam main up to its full steam-delivering capacity, and the pumping capacity varied by the number of ejectors in service.

The efficiency of the steam ejector is very low, but in recent years it has been increased, and there seems no reason why still better results could not be obtained by careful investigation and experiment. If by proper design an ejector working on a much larger scale than usual could be built that would develop 1 hp. on 50 lb. of steam, then we would have a very practical and efficient pump to use in such a contingency as occurred on the "Titanic."

A few approximate figures will give some idea of what could be accomplished along this line. The combined boilers of the "Titanic" ought to have been able to easily deliver 500,000 lb. of steam per hour, and this at 50 lb. per hp. for the ejectors would give 10,000 hp. as the pumping capacity. This is equivalent to $10{,}000 \times 550 = 5{,}500{,}000$ *ft.-lb. per sec.* If this was used in elevating the water 20 ft., then the weight of water lifted per second would be

$$5{,}500{,}000 \div 20 = 275{,}000 \text{ } lb.$$

If a cubic foot of salt water weighs 64 lb., then the discharge would be

$$275{,}000 \div 64 = 4297 \text{ } cu.ft. \text{ } per \text{ } sec.$$

Approximately what size hole in the side of the ship would it take to balance this discharge? Assume that the average center of depth of the hole is 20 ft. below the water line, and the rate of inflow at this depth to be equal to $0.8\sqrt{2g \times 20}$ which equals about 29 ft. per sec. This in all probability would be a higher velocity than actual, but errs on the safe side. As the inflow is equal to 4297 cu.ft. per sec., and as the inrushing water travels with an average velocity of 29 ft. per sec., then the area of the opening will be

$$4297 \div 29 = 148 \text{ } sq.ft.$$

say a hole 49 ft. long by 8 ft. wide. In the case of the "Titanic" this additional pumping capacity, combined with the water-tight compartments, would surely have floated the ship many more hours, if it had not actually kept it afloat and allowed collision mats to have been placed over the opening and checked the leakage until port was reached, or until a rescue ship arrived.

GEORGE P. PEARCE.

Exeter, N. H.

Engine Connected up Backwards

The article by J. H. Stratton, under this heading, in the Aug. 13 issue, I read with considerable interest.

There are some things, however, that do not look reasonable. I can understand how, by excessive compression, the pressure in the cylinder could rise above boiler pressure, but I fail to see how the engine could run when the pressure in the cylinder did not drop below boiler pressure at any time.

Perhaps Mr. Stratton can explain the phenomenon.

W. P. MARTIN.

Mt. Olive, Ill.

At first I doubted that the engine described by Mr. Stratton would run at all, but after drawing a sketch of a piston valve in several positions of stroke, I think it possible that the engine would turn over with the valve set just right. At least two statements in the article, however, need further explanation to make them clear to the average reader.

How could the cylinder show a higher pressure than the boilers except owing to compression; and if the pressure never dropped below 120 lb. (or boiler pressure), where did any power come from for compression, to say nothing of carrying the load?

If the engine ran at 650 r.p.m., it must have been a different quality of water than we have here that would flood that engine to a standstill and leave it in shape for a fresh start as soon as the water was gotten rid of. It would not seem likely that anything lighter than solid water would stop a machine that was piped up right if it was of sufficient power to run on with steam piped into the exhaust. The engines around here, especially piston-valve engines, have a way of making a bad wreck of themselves when water comes over from the boiler in large quantities.

L. JOHNSON.

Exeter, N. H.

Inward Swinging Fire Doors

The editorial, "Precautions Against Damage from Tube Failures," in the Aug. 20 issue, mentions fire-doors that swing inward as a preventive against scalding the fireman when a water-tube boiler tube blows out. This type of door is a great aid in this respect, and the editorial has not overstated its advantages.

I have experienced two cases of this kind. In one the fireman was standing directly in front of the boiler when the tube let go. The door was shut and when the excitement was over, it was found that nearly all the fire was banked up against the doors. Had the doors opened outward, the fireman would have been seriously or perhaps fatally scalded, as there was no warning and he could not have gotten away in time to save himself. The other case was nearly the same except that the fireman had just fired up and stepped to one side. The doors fitted quite tight and it was possible for the fireman to pass in front of the boiler to reach the ladder by which he reached the top of the boiler to shut off the stop valve. These boilers were not equipped with automatic stop valves as they should have been.

Although this type of door has the advantage of safety, it has some serious disadvantages (at least the kind I used had). They do not fit tight enough after the setting becomes old and warped out of shape, and to get them open wide the brick arch over the door must be set with very little spring and be higher above the opening to make room for the lining when the door is open. This leaves a part of the door frame exposed to the fire which soon cracks or warps the casing, causing the door to bind or leak air, and opening it against the arch causes the arch to become loose and fall down. I find that the life of the door and arch are not as long as that of the outward-swinging door.

It has another advantage, however, that the editorial did not mention. The fireman's side is not exposed to a red-hot door-lining while firing or cleaning fires. As they are counter-balanced, they are much easier to open than the ordinary door. In the above mentioned plant, boilers were in use, fitted with both kinds of doors, and the inward-swinging door boilers were preferred to those with the outward swing.

J. C. HAWKINS.

Hyattsville, Md.

Inquiries of General Interest

All Questions Must be Accompanied by Name and Address—Not for Publication

Temperature of Superheatea Steam

What would be the temperature of steam at 150 lb. gage pressure superheated 250 deg.?

O. S.

The temperature of steam at 150 lb. gage pressure being 366 deg. F., superheated 250 deg. the temperature would be 366 + 250 = 616 deg. F.

Increasing Height of Chimney

What would be the advantage of increasing the height of a chimney from 125 to 200 ft.?

A. C. B.

The amount of draft would be increased about as the square root of the increased height, i.e., as 11.18 to 14.14, or an increase of about 26 per cent.

Liquid Fuel Injectors

In place of using compressed air for fuel injection in Diesel engines, why not use superheated steam generated from the heat of the exhaust?

E. R.

A pressure of 500 to 900 lb. per sq.in. is required, and as the temperature of the exhaust is only about 700 deg. F. it is unsuitable for generation of steam of pressure and temperature required for fuel injection.

Use of High Vacuum

Why is a higher vacuum than 26 in. less desirable in operating reciprocating condensing engines than in operating steam turbines?

B. G.

In most reciprocating engines, increased condensation due to higher vacuum would offset gain due to expansion, unless the steam were highly superheated. In the steam turbine such losses do not occur, as high- and low-pressure steam do not alternately come in contact with the metallic surfaces.

Laying Up a Boiler

What is a good method to prevent rusting of a boiler and iron smoke connections which are to be laid up out of use for several months?

J. R.

Wash out the boiler with clean water, disconnect the feed-water pipe and after draining off the water remove the safety valve or a manhole on top of the boiler. Then with a very light fire drive out all moisture and close the boiler up tight. All flues and smoke connections should be thoroughly cleaned, the smoke uptake should be disconnected from the stack and the boiler flues, smoke flues and stack connection should be stopped up to prevent circulation of air from depositing moisture.

Hold of Expanded Tube Ends

Can expanding the tube ends of a return-tubular boiler without special flaring or beading, be depended on for safely staying the heads, though the tubes might be expanded tightly enough to prevent leakage when the boiler is under cold-water hydrostatic test pressure?

B. E.

No, as the hold of expanding is almost entirely dependent upon friction, and a cold-water test pressure is no criterion of the holding power or safety of the tube ends when the boiler is under pressure, especially after it has been subjected to changes of temperature. Serious explosions under ordinary working pressures are directly tracable to failure of whole sets of tubes that have been only expanded by being drawn through the tube-sheet holes, due to bulging of one of the heads, though they have withstood 150 per cent. hydraulic test pressure, but failed without warning under ordinary working conditions.

Effect of Back Pressure

What effect will increase of back pressure for exhaust-steam heating have on the power and economy of an engine driving the same load?

W. F. B.

As the same mean effective pressure will be required, each pound increase of average back pressure will require a pound increase of the average forward pressure. The engine will be able to drive the same load, provided the initial pressure or length of cutoff can be increased so as to obtain the same mean effective pressure. In consequence of later cutoff or higher initial pressure being required for the same load, more steam will be needed per indicated hp. Under average conditions the percentage increase of steam required will be somewhat greater than the ratio of increase of average back pressure to the mean effective pressure, though for commercial considerations, the ratios may be considered as equal. If, for instance, the required mean effective pressure is 50 lb. and it is proposed to increase the average back pressure by 2 lb., then the same increase of steam will be required as though the change consisted in an increase to 52 lb. mean effective pressure. Knowing the hp. constant of the engine for 1 lb. mean effective pressure, it is readily ascertained to what per cent. increase of power the proposed increase of back pressure would be equivalent.

Starting a Long-Idle Plant

What should an engineer attend to if placed in charge of a power plant to operate it if the plant has not been operated for two years?

H. H.

Make hammer test and if necessary renew all defective boiler connections in each instance back to the first stop valve, and see that try-cocks and water-column connections are clean of any scale or sediment and all stop, check and safety valves are in good working order. Have the steam gage tested. Clean out furnaces and back connections and remove all soot from smoke uptakes and flues and from the base of the chimney. Make internal and external inspections of boilers and settings. Examine water supply and overhaul feed pumps and injectors. Examine whether blowoff connections are open to their place of discharge. After boiler and connections are cleaned and put in safe working order, fill boiler and with a very slow wood fire examine setting and stop any air leaks and repair them. Then slowly getting up steam, note any leaks and repair them, but not while under pressure. Make a light hammer test of heater and connections.

Remove engine-cylinder head, clean out grit and dirt from all steam passages by blowing steam through them and have cylinder perfectly clean before replacing the head. Repack stuffing-boxes and inspect valve setting and all governor connections.

Trace out all engine connections and examine valve setting. Uncover and clean all bearings and flush them with clean thin oil. In starting the engine, use plenty of lubricant. Before putting steam in the engine follow up all transmission connections to see that they are in safe running order, and in starting up run slow and have the load laid on gradually. The most important considerations are to safeguard life, limb and property and conditions of plant that will insure continuous operation.

Study Questions

This Week's Questions
Last Week's Answers

(106) The armature of a shunt-wound dynamo has a resistance from brush to brush of 0.03 ohm. The voltage across the brushes is 110 volts and a current of 150 amp. flows in the external circuit. The shunt field has a resistance of 55 ohms. What is the copper loss in the armature?

(107) A vessel (like a drinking glass) 9 in. in diameter at the top, 4½ in. in diameter at the bottom and 10 in. high, is one-quarter full of water. How large a ball can be put into it and be just covered by the water?

(108) If the pointer of a vacuum gage stands at 9, what is the approximate absolute pressure in the condenser?

(109) From an indicator card the steam consumption is computed to be 28.6 lb. per i.hp. If the mechanical efficiency is 87 per cent., what is the steam consumption per brake-horsepower?

(110) What is the thermal efficiency when the initial steam pressure is 85 lb. absolute and the exhaust pressure 2 lb. above vacuum?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

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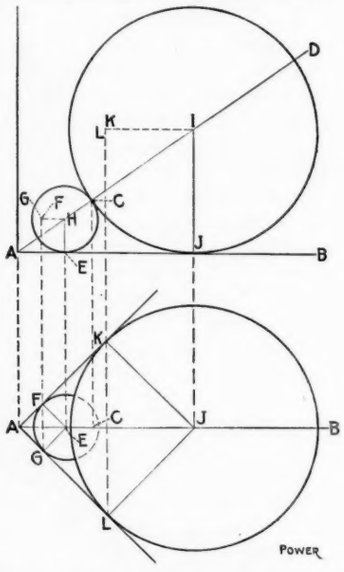
(101) The lower diagram is a top view of the two balls in the corner of the room. The line *AB* represents a vertical plane passing through the centers



of the two balls, their point of tangency *C* and the corner of the room. The upper diagram is a projection of this plane, or may be considered a vertical section through the centers of the balls; the point of contact between them *C* will lie on a line *AD* through the corner of the room and the ball centers. It is given that the diameter of the small ball is 12 in. Then its radius is 6 in. and this will be the distance from the center to the contact with either wall as *EF* and *EG*. Since *AFE* and *AGE* are right angles *AF* and *AG* will also be 6 in. long. The horizontal distance from the corner to the center of the ball *AE* will be

$$AE = \sqrt{AF^2 + FE^2} \qquad (1)$$

The actual distance will be *AH*, which is

$$AH = \sqrt{AE^2 + EH^2} \qquad (2)$$

EH also is the radius of the ball 6 in.; hence from (1) and (2)

$$AH = \sqrt{6^2 + 6^2 + 6^2} = \sqrt{108} = 10.392$$

The distance *AC* (upper diagram) is *AH* + *HC*, or

$$AC = 10.392 + 6 = 16.392 \text{ in.}$$

The radius of the larger ball is to be determined. Call it x. Then the distance from the corner to the center of the larger ball ($AC + CI$) is

$$AI = 16.392 + x \qquad (3)$$

AI can also be found, as was *AH*, by solving right-angled triangles.

$$KJ = x = AK$$

Then

$$AJ = \sqrt{KJ^2 + AK^2} = \sqrt{x^2 + x^2}$$

and

$$AI = \sqrt{AJ^2 + IJ^2} = \sqrt{x^2 + x^2 + x^2}$$
$$= \sqrt{3x^2} = x\sqrt{3} \qquad (4)$$

Equating (3) and (4)

$$16.392 + x = x\sqrt{3}$$
$$x\sqrt{3} - x = 16.392$$
$$x(\sqrt{3} - 1) = 16.392$$
$$x = \frac{16.392}{0.732} = 22.393$$

The diameter of the larger ball is therefore

$$22.393 \times 2 = 44.786$$

or very closely 44¾ in.

(102) The man's path will be the diameter of the circle; the automobile's, its circumference. The distance the man goes will therefore be as *D* and the distance the automobile goes as $\frac{3\pi D}{2}$ in the same time; hence the automobile must go

$$\frac{\frac{3\pi D}{2}}{D} = \frac{3\pi}{2} = 4.7124$$

times as fast as the man. If the man goes eight miles an hour the automobile must go

$$8 \times 4.7124 = 37.699$$

or approximately 37.7 miles an hour.

(103) J. B. Stanwood's formula as given in "Kent's Pocketbook" is

$$W = \frac{700{,}000\, d^2 s}{D^2 R^2}$$

where

d = Diameter of cylinder, in. = 18;
s = Stroke, in. = 36;
D = Diameter of flywheel, ft. = 15;
R = R.p.m. of flywheel = 90;
W = Weight of rim, lb. = ?

Substituting

$$W = \frac{700{,}000 \times 18^2 \times 36}{15^2 \times 90^2} =$$

$$\frac{700{,}000 \times 324 \times 36}{225 \times 8100} = 4480 \text{ lb.}$$

(104) The theoretical mean effective pressure will be the same as admission 110 lb. From the formula

$$I.hp. = \frac{plan}{33{,}000}$$

where

p = Mean effective pressure, lb. = 110;
l = Length of stroke, ft. = $\frac{30}{12}$ = 2.5 ft.;
a = Area of cylinder = 0.7854 × 18^2 = 254.47 sq.in.;
n = Strokes per min. = r.p.m. × 2 = 150.

$$I.hp. = \frac{110 \times 2.5 \times 254.47 \times 150}{33{,}000} =$$

$$\frac{10{,}496{,}887.5}{33{,}000} = 318.0875 \text{ hp.}$$

(105) The total heat in 1 lb. of steam at 11 lb. abs. (from steam tables) is 1144.9 B.t.u. Leaving at 120 deg. F. the 1 lb. of steam still contains 120 — 32 = 88 B.t.u. It has, therefore, lost

1144.9 — 88 = 1056.9 *B.t.u.*

The cooling water has been raised

110 — 53 = 57 deg.

Then each pound of it has absorbed that number of B.t.u. The quantity of heat lost by each pound of steam is therefore sufficient to raise

$$\frac{1056.9}{57} = 18.54 \text{ lb.}$$

of water through the range specified. Since the engine uses 26 lb. of steam per horsepower-hour and runs at 53 hp., the condenser receives

26 × 53 = 1378 *lb. per hr.*

The cooling water required per hour is therefore

18.54 × 1378 = 25,548.12 *lb.*

The formula for calculating the surface of condensers, as given by H. L. Hepburn in POWER, April, 1902, is

$$AU = S\frac{H - N}{F - I} \times log.e\frac{T_s - I}{T_s - F}$$

where

A = Area of surface, sq.ft.;
U = B.t.u. transmitted per sq.ft. per hour per deg. mean difference of temperature between the steam and water = 192;
S = Pounds of steam per hr. = 1378;
H = B.t.u. above 32 deg. F. in 1 lb. of steam = 1144.9;
N = B.t.u. in 1 lb. of condensed steam = 88;
F = Final temperature of the water = 110 deg. F.;

I = Initial temperature of the water = 53 deg. F.;

T_s = Temperature of the steam.

From steam tables T_s = 197.75 deg. F. Substituting in the formula

$$A \times 192 = 1378 \frac{1144.9 - 88}{110 - 53} \times log.e \frac{197.75 - 53}{197.75 - 110}$$

$$A \times 192 = 1378 \frac{1056.9}{57} \times log.e \frac{144.75}{87.75}$$

$$A \times 192 = 25{,}551.02 \times log.e\ 1.65$$

$$A = 133.06 \times log.e\ 1.65$$

Log.e of 1.65 (from tables of hyperbolic logarithms) is 0.5008. Then

$A = 133.08 \times 0.5008 = 66.646$ *sq.ft.*

Free Electrical Lectures in Brooklyn

The Brooklyn Institute of Arts and Sciences announces a series of five lectures by Dr. A. E. Kennelly, of Harvard University, on the elements of hyperbolic functions and their applications to electrical engineering. The lectures, which will be illustrated by lantern slides, are free to the profession and will be given on the second Thursday evening of each month of the season in the physics lecture room of the Polytechnic Institute, Brooklyn, at 8:15. A working acquaintance with ordinary circular functions of plane trigonometry will be assumed, but no calculus or mathematics above algebra will be involved.

The dates and subjects of the lectures are as follows: Oct. 10, hyperbolic angles: their properties and applications; Nov. 14, the behavior of direct-current lines of uniform linear conductor resistance and dielectric conductance, in the steady state, as simplified by the use of hyperbolic functions; artificial lines and equivalent circuits; Dec. 12, the behavior of alternating-current lines of uniform linear conductor-impedance and dielectric admittance, in the steady state, and also in simple unsteady states, as simplified by the use of hyperbolic functions; Jan. 9, the application of hyperbolic functions to long alternating-current power-transmission lines; Feb. 13, the application of hyperbolic functions to telephone lines, loaded and unloaded, single and composite.

Standardizing Electricity in France

A bill for the protection of consumers against fraud is reported in preparation by the French Minister of Commerce. It will establish standards for electricity, heat and candlepower and the service companies will be compelled to conform to its standards. Meters and lamps will be subject to the same regulations as weights and measures.

Gas Engine Valve Pliers

A new tool, which seems to fill a want for users of four-cycle engines, is shown in Fig. 1. Fig. 2 shows its application

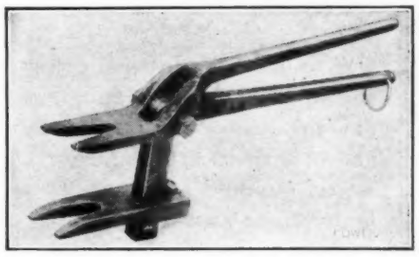


FIG. 1. VALVE PLIERS

to a poppet valve of a gas or gasoline engine. Valves of this class show very little wear if kept clean, and with this tool but a few moments is required to remove a valve and clean it.

The ring on the handle of the pliers is

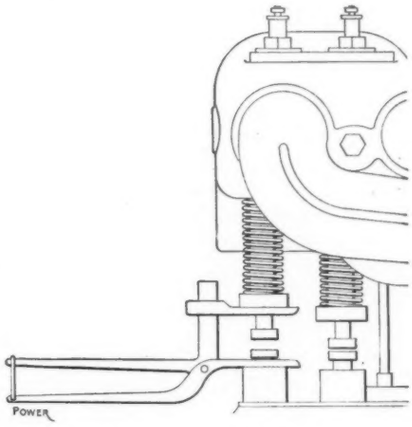


FIG. 2. APPLICATION OF PLIERS

for locking them when the spring is collapsed. The engaging jaws are forked to give them a wide range in valve stem size. The pliers are made of steel case-hardened drop forging by C. A. Cummings, P. O. Box 373, Canton, Ohio.

100,000,000 Gallon Turbine Driven Pump

A steam turbine-driven centrifugal pump of 100,000,000 gal. per day capacity is to be installed in the Ross pumping station of the Pittsburgh Water Works by the DeLaval Steam Turbine Co., of Trenton, N. J. This pump will be the largest steam turbine-driven centrifugal pump ever built in this country, the rated capacity of 100,000,000 gal. per day against a total head of 56 ft., amounting to over 980 water horsepower. The pump is guaranteed to show a duty of 115,000,000 ft.-lb. per 1000 lb. of dry steam, all steam used by the condensing equipment being charged to the main unit.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

Blink Boffum, of the Waters power plant, opines that the Nifetown *Blade* is too cutting in remarking that he is "a rising young man." Says he was willing to stay down, but "ther dum ingineer weighted ther safety, an' when ther old biler riz," Blink had no choice.

"Anxious Subscriber" inclosed $2 and asked for the solution of a perpetual-motion problem. The best we could do to relieve his anxiety was to return the $2. As for perpetual motion, "there ain't no sich thing," as the farmer said the first time he saw a giraffe.

The following is an appreciation of our humble efforts to drive away dull care and make merry in our travail:

deer mister Spilway. Them 2 lines of yourn beets awl for testing my Mans sobriarty. he cum hoam 2 ours lait from the plant 1 nite, sum werce for ware. "I bin makin a test, Malviny," sez he. "Justus Jinkins," sez i, "ef you kin repeet them 2 lines from yure Power, 'just jests, jabs, joshes and jumbils', ile allow yure tung aint mixt ez bad as yure laigs be."

Justus aint agivin awl the datter of my test, but i heerd him tell Ezry Skidmore that he was rite het up ovar the konsidabul low pursentige of his fishency, whatevar them things is. hes goin ter vote along with the "drys" come this fall.

Yure obleeged friend,
MALVINA JINKINS.

At a Pittsburgh theater recently the orchestra played "Meet Me in the Shadows" when an accident in the lighting plant shut off the current and left the audience in darkness. American humor, when it is not too dark to see the joke, has often prevented a panic.

An ingenious Richmond is in the electrical field with a scheme for putting people to sleep by electricity. The lure of Morpheus is good enough for us at night-time. Keeping the wolf from the door makes us sufficiently alert and wakeful day-times.

This engineer made good, boasts a contemporary. Bully! With a few added details, the announcement would make an excellent construction note.

Looks like the New York police force was in need of a few lightning-arresters as some of them have been under heavy "charges" recently. A few might well be transformed and stepped down—and out.

Steam Boiler Explosions*

By William H. Boehm

In which it is pointed out that the principal causes of boiler explosions are due to faulty design and construction, false factor of safety, improper care and other faults due to ignorant operation.

Every year there occur in the United States between 1300 and 1400 serious boiler accidents, of which 300 to 400 are violent explosions. These accidents kill between 400 and 500 persons, injure 700 to 800 more, and destroy more than a half million dollars worth of property. These disasters have but scant respect for boiler types—for they occur with water-tube boilers, although with them violent explosions occur less frequently than with fire-tube boilers. They also occur with low-pressure boilers.

Design and Construction

It is of the utmost importance that boilers be carefully designed, that the stresses to which they are subjected be accurately computed, that suitable material be specified, that the material be critically examined for flaws or defects, that specimens of the material be tested to determine its strength, that no abuse of the material be allowed in the process of constructing the boiler, and that the completed boiler be subjected to a thorough inspection and a hydrostatic test before being put into service.

Stress in Girth and Side Seams

The stress in the girth seams of a boiler may be obtained by the formula

$$S = \frac{rp}{2t}$$

in which

S = Stress per square inch to which the material is subjected;
r = Radius of the boiler;
p = Steam pressure carried;
t = Thickness of the shell.

The formula means that if we multiply the shell radius by the steam pressure, and divide the product by twice the shell thickness, the result will be the stress in pounds per square inch to which the material in the girth seams is subjected.

The stress in a side seam of a boiler may be obtained by the formula

$$S = \frac{rp}{t}$$

which means that if we multiply the shell radius by the steam pressure and then divide the product by the shell thickness, the result will be the stress in pounds per square inch to which the material in a side seam is subjected.

An inspection of these formulas show that the stress in a side seam is just twice as great as the stress in a girth seam. It is for this reason that the side seams are usually double riveted when the girth seams are only single riveted.

Bursting Pressure

The pressure required to rupture the shell of a cylindrical boiler is given by the formula

*Excerpt from a lecture delivered at Cornell University before the state branch of the American Society of Mechanical Engineers.

$$P = \frac{st}{r}e$$

in which

P = Bursting pressure in pounds per square inch;
s = Tensile strength of the material in the boiler;
r = Radius of the shell;
e = Efficiency of the riveted side seam.

Stated in words the formula means that if we multiply together the tensile strength of the material, the thickness of the shell, and the efficiency of the riveted side seams, and then divide the product so obtained by the radius of the shell the result will be the steam pressure at which explosion will occur.

If it so happens that the efficiency of the girth seam is too low by reason of improper design, then the girth seam may fail instead of the side seam, in which case the bursting pressure is given by the formula

$$P = \frac{2ts}{r}e'$$

which means that if we multiply together twice the thickness of the shell, the tensile strength of the material, and the efficiency (e') of the girth seam, the result will be the steam pressure at which explosion will occur. The girth seam is not likely to fail before the side seam, because to do so the efficiency of the girth seam would have to be less than half that of the side seam—a weakness that should not exist in a boiler of proper design and construction.

Factors of Safety

The formulas expressed herein give the pressure at which the boiler will explode and not the pressure at which it may be safely operated. It is usual in boiler practice to fix the allowable working pressure for a new boiler at one-fifth the computed bursting pressure and to decrease the pressure allowance as the age of the boiler increases.

This is equivalent to saying that the factor of safety applied to a new boiler is usually not less than five. The term "factor of safety," is often misunderstood and a better name would be "factor of ignorance," as it is as much a factor of ignorance as it is a factor of safety.

Take, for example, the case of a new boiler operated with the safety valve set at 100 lb. If the computed bursting pressure be 500 lb., then the assumed factor of safety is five. The assumption, however, is based upon the tensile strength stamped in the plate by the maker and this strength is only true of that particular part of the plate from which the test specimen was cut and not necessarily true of any other part.

As a matter of fact it is current practice to cut the test specimen from the outer edge of the plate and the strength there is almost invariably greater than the strength at the center of the plate. The reason is that after liquid steel is poured into a mold its solidification in forming an ingot proceeds in much the same manner as does the solidification of water in forming a block of ice. That is to say, the impurities and gases are driven toward the center.

Boiler plates made by the rolling of such an ingot will, therefore, have more impurities and less strength at the center of the plate than at the outer edges, and this variation in strength is very considerable. Then, too, the stress at which the elastic limit of the material is reached is little more than half the stress at which rupture occurs.

Besides our ignorance of the dependable strength in all parts of the plate, there is also our ignorance of the character of the workmanship in the boiler. We cannot be certain that all rivet holes come fair, or that incipient cracks have not been set up by an abuse of the material during the process of construction. It is seen, therefore, that factors of safety are really made up of two parts—one part a true factor of safety, the other a pure factor of ignorance.

Causes of Boiler Explosions

Boiler explosions may be attributed to improper construction, improper installation, or incompetent or careless operation.

Improper construction may consist: of unsuitable or inferior material; poor workmanship; abuse of material, as when unmatched rivet holes are drift-pinned to place, or uncylindrical shells are sledged to form; of employing the more dangerous lap joint for the side seams instead of the more safe and more sensible butt joint, etc.

The lap joint is dangerous because this form of construction promotes the formation of incipient cracks in the upper surface of the lower lap where they may

be impossible of detection. These cracks extend from rivet hole to rivet hole and gradually deepen with the continued raising and lowering of the steam pressure until the metal, no longer capable of resistance, gives way and causes a violent explosion.

The lap joint is given the preference over the butt joint solely, because, it appears at first thought to be cheaper. But there is no excuse for its existence, and its employment in the construction of new boilers should be prohibited by law in all states, as it now is in some.

Improper installation may consist of so supporting the boiler and its piping as to allow temperature changes to set up dangerous stresses in the material, of improperly attaching the usual appurtenances such as safety valves, steam and water gages, check, blowoff, stop valves, etc.

Incompetent or careless operation may consist in allowing the steam gage to get out of order, in allowing the water-gage connections to become so clogged as to indicate ample water when there is none in the boiler, in allowing the safety valve to become so stuck to its seat as to fail to blow at the pressure for which it was set, in allowing grease to enter or scale to accumulate in the boiler, in allowing large quantities of cold water to be impinged against hot plates, in allowing the water to be driven from the heated surfaces by forced firing, in allowing a large valve to be opened too suddenly, in allowing two boilers to be cut in on the same steam main when their pressures are unequal, and in allowing minor repairs to be neglected until they endanger the whole structure.

It is significant that many violent boiler explosions occur either just prior to the starting of the engines in the morning, or while they are idle at the noon hour, or shortly after they have been shut down for the day. One reason is that when steam is not being drawn from the boiler it accumulates rapidly; and if the safety valve fails to relieve the pressure, explosion soon follows.

The rapidity with which the bursting pressure is reached may be shown as follows:

Let

T = Time in minutes required to reach the bursting pressure;
W = Weight of water in the boiler;
t = Temperature of the steam at bursting pressure;
t' = Temperature of the steam at normal working pressure;
U = Number of heat units per minute supplied by the furnace and absorbed by the water.

The heat balance is then represented by the equation:

$$UT = W\ (t - t')$$

$$T = \frac{W}{U}(t - t')$$

which means that if we multiply the difference between the temperature of the steam at bursting pressure and at normal pressure by the weight of the water in the boiler, and then divide the product by the number of heat units supplied per minute by the furnace, the result will be the number of minutes that will elapse from the time the openings are all closed until explosion follows.

Take, for example, a 100-hp. boiler containing at normal level 5000 lb. of water and suppose it uses 50,000 heat units per minute when evaporating 50 lb. of water per minute. Then if the normal gage pressure be 85 lb., the corresponding temperature of the steam is 327 deg., and if the bursting gage pressure be 485 lb., the corresponding temperature of the steam is 467 deg.; and the time required to reach the bursting pressure with all steam openings closed and the safety valve stuck is:

$$T = \frac{5000}{50{,}000}(467 - 327) = 14\ min.$$

That is, with a stuck safety valve, only 14 min. would elapse from the time the engines were shut down until the explosion followed.

Explosive Energy in Heated Water

The temperature of the water in a boiler is approximately the same as the temperature of the steam with which it is in contact. If the fire be drawn when the openings are closed, ebullition ceases. If a valve be opened, ebullition starts again, even though there still be no fire under the boiler.

It is plain, therefore, that with the openings closed it is the pressure on the surface of the water that prevents further generation of steam. It is also plain that if a small rupture occurs below the water line a violent explosion may not ensue. But it ought to be evident that if a large outlet above the water line be suddenly opened, as, for example, when a steam pipe fails, then the sudden liberation of the pressure on the surface of the high-temperature water will allow it to flash suddenly into steam and cause a violent explosion and water-hammer that will disrupt the strongest possible construction.

Grease in Boilers

Grease does not dissolve or decompose in water, nor does it remain on the surface. Heat in the water and its violent ebullition causes the grease to form in sticky drops which adhere to and varnish the metal surfaces of the boiler. This varnish by preventing the water from coming into intimate contact with the metal, prevents the water from absorbing the heat, and this causes a blistering or burning of the plate that often results in a serious rupture, or a violent explosion.

Scale in Boilers

If scale is allowed to accumulate to any considerable thickness in a boiler, a bag or rupture of the shell is inevitable, unless the scale happens to be of a spongy formation, which is not often the case. Just why this is so, is shown by the following simple experiment.

Take an ordinary granite iron or tinned iron stewpan and firmly glue to its underside a postage stamp. Pour water into the pan and place it on a gas stove so that the postage stamp will be in direct contact with the flame. Leave the pan on the stove until the water has boiled violently and then examine the stamp. The stamp will not even be charred, much less burned, notwithstanding that it was on the underside of the pan and in direct contact with the hottest part of the flame.

Now put into the pan a mixture of water and portland cement half an inch thick. This, when set, will be the equivalent of half an inch of scale. Repeat the experiment made before and it will be found that the stamp will burn up very quickly.

The reason that the postage stamp is not charred by the flame when no scale is present is that the water, being in immediate contact with the thin bottom of the vessel, absorbs the heat as fast as it is put into the vessel by the flame. The result is that, no matter how hot the flame may be, the bottom of the vessel remains at practically the same temperature as the boiling water with which it is in contact. In an open vessel the temperature of boiling water is 212 deg. and this is not sufficiently high to char paper. When scale is present, the water cannot absorb the heat as fast as it is put into the vessel by the flame, and as a result the temperature becomes greater than 212 deg. and burns the postage stamp.

It is the same with steam boilers. If the water comes in direct contact with the thin plates, the heat is absorbed, the temperature of the plates remains practically the same as the water, and no harm is done. If there be a considerable thickness of impervious scale in the boiler, the water cannot absorb the heat as fast as it is put into the plates by the furnace, and so the plates become overheated, get red, become plastic, and finally give way to the force of steam pressure, causing a bag, or a rupture, or a violent explosion of the boiler.

Scale endangers the safety of boilers in other ways. It clogs the feed pipes, preventing the feed water from freely entering the boiler. It clogs the connections to the water gage, causing it to indicate ample water when it is at a low level in the boiler. Pieces get under valves and prevent their closure.

Scale in boilers is a serious matter, and in order to prevent its accumulation, it is good practice to eliminate the scale-

forming matter from the feed water before allowing it to enter the boiler. This can be accomplished either mechanically by means of separators, or chemically by treating the water in vats especially arranged for the purpose. If preferred, compound may be fed with the water into the boiler, but in such case the water should be analyzed, and the proper compound prescribed by a chemist making a specialty of such matters. Kerosene fed into the boiler has proved beneficial in many instances.

INSPECTION AND INSURANCE

It is an almost universal custom for boiler owners to have their boilers insured and inspected. The insurance serves as a guarantee that the inspections will be intelligently and carefully made and the inspections lessen the chance of accident.

When boiler insurance is carried, an inspector visits the plant at regular intervals and critically examines the boilers, both internally and externally. During the past 10 years the company represented by Mr. Boehm made 1,101,140 examinations and reported 140,989 defects, many of which consisted of dangerous fractures in or near the riveted seams, and that one boiler out of every eight examined, contained defects serious enough to warrant their being reported.

Power Possibilities of British Columbia

The work of investigating the power possibilities of British Columbia, inaugurated last year, has been resumed under the direction of Arthur V. White, of the Canadian Commission of Conservation, who has charge of all work of this nature in western Canada. Three engineering parties are now engaged examining the Fraser River and tributaries between Lytton and Fort George. This embraces one of the largest drainage basins on the continent.

The recent report on the water powers of Canada drew special attention to the fact that the water powers available for development from an economic standpoint are fewer in number and potentiality than is popularly supposed. Their findings have an important bearing on the question of conserving these powers in the public interest in view of the new towns and cities springing into existence in British Columbia.

In the report the commission promised that special investigations would be carried on in the prairie provinces and British Columbia, because all the required data were not available at the time of publication. The water powers of the Kootenay River have already a profound economic influence in the Kootenays and Boundary district. Similarly the same is expected to be effected by the Adams River water power throughout the Shuswap Lake district and South Thompson River county, where irrigation is being resorted to.

Reported Boiler Explosion

According to a newspaper report a boiler explosion at the South Madison St. plant of the Bloomington, Ill., Railway, Electric & Heating Co., at 6:35 p.m., Sept. 21, partly wrecked the building, injuring one fireman, but not fatally. The entire service was disabled.

The No. 12 boiler, which is said to have exploded, had been out of service for six weeks for general repairs, which were done by the plant workmen. It had been in service but 20 min., when the accident occurred. No reason is given as to the probable cause.

Artificial Graphite

The manufacture of artificial graphite, according to a bulletin of the U. S. Geological Survey, is conducted by means of the electric furnace, an anthracite coal carrying small amounts of evenly distributed impurities being the material from which the ordinary grades are made.

For obtaining the purest grades of graphite, petroleum coke is substituted for anthracite. The process for the manufacture of graphite was patented in 1896 and its commercial development has been so rapid that at present the output of artificial graphite in the United States is greater than the whole domestic production of natural crystalline graphite.

Testing the British Yard Measure

The ceremony of testing the British yard measure and the British pound weight which occurs every 20 years is soon to be performed. A copy of each is buried in a wall of the House of Commons, and is compared at stated intervals with the originals to see if the latter have undergone any deterioration. They order these things better in France, where, by the skill of an American mechanic and the science of an American professor the length of the meter has been determined in terms of light waves so that it could at any time be recovered if the original standard were lost. This is more accurate than the original determination in terms of the circumference of the earth, which errs a little through defective geographical measurement.—*Popular Engineer.*

The Union Electric Light & Power Co., St .Louis, Mo., on Sept. 27 took a party of 150 manufacturers as its guests to Keokuk, Iowa, to inspect the hydro-electric plant under construction at that point by the Mississippi River Power Co., which will supply much of the power to be used in St. Louis.

NEW PUBLICATION

THE LOCOMOTIVE. Published by the Hartford Steam Boiler & Inspection Co., Hartford, Conn. 1910-1911. Size, 5½x9 in.; 254 pages; illustrated; complimentary.

This book is compiled from the quarterly bulletins published by the company. The volume is filled largely with data relating to boiler explosion and boiler inspection. There are also articles treating with the care of boilers.

One article in particular dealing with the management and care of steam boilers is of more than passing interest to those who are not thoroughly familiar with this phase of steam-plant operation. The instructions given therein are based on sound logic and common sense.

The list of boiler explosions during the two years covered by the book give food for thought to those who may feel secure in the belief that boiler explosions are not common.

There are also many short instructive and interesting articles too numerous to mention.

BOOKS RECEIVED

DIESEL ENGINES. By A. P. Chalkley. D. Van Nostrand Co., New York. Cloth; 226 pages, 5¼x8½ in.; illustrated; plates; tables. Price, $3.

THE PRINCIPLES OF PARALLEL PROJECTING-LINE DRAWING. By A. A. Adler. D. Van Nostrand Co., New York. Cloth; 66 pages, 5½x8¾ in.; 48 illustrations. Price, $1.

BOILER DRAUGHT. By H. K. Pratt. D. Van Nostrand Co., New York. Cloth; 138 pages 4¾x7½ in.; 29 illustrations; tables. Price, $1.25.

GAS AND OIL ENGINES. By Alfred Kirschke. D. Van Nostrand Co., New York. Cloth; 159 pages, 4½x7¼ in.; 55 illustrations; tables. Price, $1.25.

MATERIALS AND CONSTRUCTION. By James A. Pratt. Blakiston's Son & Co., Philadelphia, Penn. Cloth; 196 pages, 4⅜x7 in.; 86 illustrations; tables. Price, 90c.

ENERGY AND VELOCITY DIAGRAMS OF LARGE GAS ENGINES. By Paul L. Joslyn. The Gas Engine Publishing Co., Cincinnati, Ohio. Cloth; 6x9 in.; 62 diagrams. Price, $2.

PERSONAL

Charles H. Clark, formerly chief engineer of the Connersville (Ind.) Light, Heat & Power Co., has been appointed manager of the Martinsville (Ind.) Light, Heat, Power & Water Co.

N. A. Carle has been recently appointed chief engineer of the Public Service Electric Co., with headquarters at Newark, N. J. Mr. Carle is a member of the American Society of Mechanical Engineers, the American Society of Civil Engineers and the American Institute of Electrical Engineers.